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Generic Issue 87: Flexible Wedge Gate Valve Test Program

Phase II Results and Analysis

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Prepared for
U.S. Nuclear Regulatory Commission

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ABSTRACT

Qualification and flow isolation tests were conducted to analyze the ability of selected boiling water reactor process valves to perform their containment isolation functions at high energy pipe break conditions and other more normal flow conditions. Numerous parameters were measured to assess valve and motor-operator performance at various valve loadings and to assess industry practices for predicting valve and motor operator requirements. The valves tested were representative of those used in reactor water cleanup systems in boiling water reactors and those used in boiling water reactor high-pressure coolant injection (HPCI) steam lines. These tests will provide further information for the U.S. Nuclear Regulatory Commission Generic Issue-87, "Failure of the HPCI Steam Line Without Isolation," and Generic Letter 89-10, "Safety-Related Motor Operated Valve Testing and Surveillance."

Physical inspection of the valves indicated that these valves, operating at design basis conditions, were very near their physical fragility limits. The excessive bearing pressure between the disc and the body guide materials resulted in yielding, spalling, and gouging of the surfaces. In some of the designs, the guide clearances were large enough to allow the disc to tilt during closure, resulting in significant damage to the sealing surfaces. Review of the data indicated that all six valves required more force during closing against high pressure flow loads than would be calculated using the standard variables in the industry's motor-operator sizing equations. The test results also indicated that the stem force equation used by industry for sizing motor operators is incomplete.

A6857-Equipment Operability
B5529-Failure of HPCI Steam Line Without Isolation (GI-87)

EXECUTIVE SUMMARY

The U.S. Nuclear Regulatory Commission (NRC) is sponsoring valve and motor-operator functionality research at the Idaho National Engineering Laboratory (INEL). Among the objectives of this research program is a task to determine what factors affect the performance of motor-operated gate valves and to determine how well industry's analytic tools predict that performance. These tasks support the NRC's effort regarding Generic Issue (GI)-87 "Failure of [High-Pressure Coolant Injection] HPCI Steamline Without Isolation." GI-87 covers three boiling water reactor (BWR) process lines: the HPCI turbine steam supply line, the reactor core isolation cooling (RCIC) turbine steam supply line, and the reactor water cleanup (RWCU) process line. All three of the BWR process lines communicate with the primary system, pass through containment, and normally have open isolation valves. The concern with the isolation valves is whether they will close in the event of a pipe break outside of the containment. A high energy steam or hot water release in the auxiliary building could result in common cause failure of other components necessary to mitigate the accident.

This research program also supports the implementation of Generic Letter (GL) 89-10 "Safety-related Motor Operated Valve Testing and Surveillance," which is applicable to all light water reactor (LWR) safety-related motor-operated valves (MOVs) as well as selected position changeable MOVs in safety-related systems.

One of the major parts of the research program was two full-scale flexible wedge gate valve qualification and flow interruption test programs, Phase I and II. The Phase II program was performed in the late summer and early fall of 1989 at the Kraftwerk Union (KWU) facilities near Frankfurt in the Federal Republic of Germany. Six valves were tested, three 6-in. isolation valves typical of those used in RWCU applications and three 10-in. valves typical of those used in the HPCI applications. One of the 6-in. valves was also tested at RCIC test conditions. In all, seventeen flow interruption tests were performed, seven of these at design basis conditions.

Two RWCU valves were tested during the earlier Phase I Test Program. As a result of that work, we expected that the valves would require more stem force to close than industry normally would have predicted. Therefore, for the Phase II Program, we set the motor-operator control switches at higher-than-normal torque values to ensure valve closure and then determine the strengths or weaknesses of a given valve design from the recorded data.

The test results clearly showed that for the GI-87 concerns, all valves that were subjected to design basis flow interruption tests required more torque and subsequently more stem force to close than would be predicted using the standard industry motor-operator sizing equation for disc load calculations with a common coefficient of friction. The highest loads recorded were the result of internal valve damage caused from the high differential pressure loads across the valve disc as it attempted to isolate flow.

The high loads encountered during the test series raise the concern that some valves installed in nuclear power plants may not have large enough motor operators to ensure closure in the event of a design basis accident. A pipe break in one of the GI-87 systems (outside containment) is a low probability but high consequence event.

The study into the phenomena affecting the stem loads in a motor-operated gate valve continues. However, the results to date indicate that the phenomena taking place inside the gate valve are much more complex than previously thought. The actual disc factor is much higher than previously believed, but this higher disc factor can be moderated for some valve applications once the self-closing force balance on the valve disc is understood.

ACKNOWLEDGEMENTS

The efforts of many people and organizations were required to successfully conduct the Generic Issue-87 test programs. For the Phase II portion of the research reported herein, we wish to thank Dr. U. Simon and W. Schoeder from Kraftwerk Union Karlstein; H. Knuedler and M. J. Schmidt from the Bechtel Kraftwerk Union Alliance; and S. B. Wilchek, T. Branum, and L. Thompson from General Physics. Dr. G. H. Weidenhamer and Dr. Roy Woods from the U.S. Nuclear Regulatory Commission assisted both in the planning and in the actual performance of the test program. We would also like to thank P. Schwieder from the Idaho National Engineering Laboratory for his help in selecting our diagnostic equipment options and his programming efforts, which decreased the data reduction time, and M. J. Russell for his insights and assistance in the interpretation of data.

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ACRONYMS AND ABBREVIATIONS

ANSI	American National Standards Institute
ASME	American Society of Mechanical Engineers
BWR	boiling water reactor
CEGB	Central Electric Generating Board
DAS	data acquisition system
EG&G Idaho	EG&G Idaho, Inc.
EPRI	Electrical Power Research Institute
GI	generic issue
GL	Generic Letter
HPCI	high-pressure coolant injection
IBM	International Business Machines Corporation
IE	Office of Inspection and Enforcement
INEL	Idaho National Engineering Laboratory
KWU	Kraftwerk Union
LVDT	linear variable displacement transformer
LWR	light water reactor
MOV	motor-operated valve
NOP/NOT	normal operating pressure and temperature
NRC	U.S. Nuclear Regulatory Commission
PC	personal computer
RCIC	reactor core isolation cooling
RWCU	reactor water cleanup

GENERIC ISSUE 87, FLEXIBLE WEDGE GATE VALVE TEST PROGRAM, PHASE II RESULTS AND ANALYSIS

1. INTRODUCTION

The Idaho National Engineering Laboratory (INEL), under the sponsorship of the U.S. Nuclear Regulatory Commission (NRC), is performing research[a] to resolve specific generic issues and develop and improve industry mechanical equipment qualification and operating and maintenance consensus standards. This overall research effort includes a program that tested the operability (opening and closing) of six full-scale motor-operated gate valves typical of the containment isolation valves installed in boiling water reactor (BWR) reactor water cleanup (RWCU) process lines and high-pressure coolant injection (HPCI) turbine steam supply lines. The valves were qualified and parametrically tested above, at, and below the pressures, temperatures, and flow conditions of a worst-case downstream pipe break in the RWCU and HPCI turbine supply lines outside of containment. One of the RWCU valves also was tested with steam to provide insights for the reactor core isolation cooling (RCIC) turbine steam supply line isolation valves.

The purpose of the test program was to provide technical input for the NRC effort regarding Generic Issue (GI)-87, "Failure of the HPCI Steam Line Without Isolation." GI-87 also applies to the RCIC and RWCU isolation valves. All three of the GI-87 BWR process lines communicate with the primary system, pass through containment, and normally have open containment isolation valves. The concern with these containment isolation valves is whether they will close in the event of a pipe break outside of containment. A high energy steam or water release in the auxiliary building could result in common cause failure of other components necessary to mitigate the accident. The test program also provides information applicable to the implementation of Generic Letter (GL) 89-10, "Safety-Related Motor Operated Valve Testing and Surveillance," for all light water reactor (LWR) safety-related motor-operated valves (MOVs) and selected position-changeable MOVs in safety-related systems.

The analyses performed to date on the measured data obtained during the GI-87 Valve Test Program and the conclusions derived from the analyses are discussed in this report. A complete analysis of the data required by the program objectives will follow in a later report. We are issuing this report now because the findings to date contain information that may be beneficial to the industry when responding to current regulatory recommendations. Additionally, the Phase II testing did not answer all of the test objectives because we could not locate a dc motor in time for the test program. That work and the open questions regarding stem factor and rate of loading will be performed on the INEL valve load simulator.

For those individuals or organizations who wish to do their own analysis of the data, the measured data are available from the NRC Public Document Room. The Phase I program is reported in *BWR Reactor Water Cleanup System Flexible Wedge Gate Isolation Valve Qualification and High Energy Flow Interruption Test* [EG&G Idaho, Inc. (EG&G Idaho), 1989] (NUREG/CR-5406). The Phase II Test Program actual measured data are reported in *Generic Issue-87 Flexible Wedge Gate Valve Test Program Phase II Data Report*, (EG&G Idaho, 1990). The Phase II data are also available in International Business Machines Corporation (IBM) personal computer (PC)-compatible format. There is also a video tape documenting the post

a. Work supported by the U.S. Nuclear Regulatory Commission, Division of Engineering, Office of Nuclear Regulatory Research, under U.S. Department of Energy Contract No. DE-AC07-76ID01570.

test disassembly and inspection of the test valves. Both the video and the magnetic data can be obtained through the INEL Technology Transfer Office at (208) 526-6042.

1.1 Background

Two sets of experiments have been conducted for the GI-87 research program. In Phase I, two full-scale RWCU valves typical of those in operating plants were tested with high energy water. The results of the Phase I program, as previously stated, were reported in NUREG/CR-5406. These results challenged the validity of many gate valve design rules. The two-valve sample was small and one valve sustained damage as a result of the high flow loading. Because of this, some experts in the industry did not accept the results as having general applicability. In the Phase II program, the NRC increased the valve sample size to six valves representative of those installed in both the HPCI and RWCU Systems.

Phase II test program objectives were reevaluated and modified to include the lessons learned in Phase I and to include tests on the RWCU valves that would be applicable to RCIC valves. The resulting objectives were as follows:

- Determine the valve stem force required to close typical RWCU, RCIC, and HPCI system isolation valves at typical operating conditions and under blowdown conditions

- Compare valve closing loads to opening loads at various system conditions

- Evaluate valve closure force components, such as disc friction, packing drag, stem rejection load, and fluid dynamic effects

- Measure the effects of temperature, pressure, and valve design on valve closing and opening loads

- Evaluate the terms and variables in the present standard valve and motor-operator sizing equations

- Provide detailed information to assist in the NRC effort regarding GI-87

- Correlate the data to develop a methodology for in-situ motor-operated valve testing, supporting the implementation of GL 89-10.

The results of the testing may also contribute to specific guidelines being developed to improve valve qualification and in-plant test standards such as American National Standards Institute (ANSI)/American Society of Mechanical Engineers (ASME) standards B16.41 (1983), OM-8, and OM-10.

Surveys of utility installations performed before the Phase I program determined that in the BWR systems of interest, the flexwedge gate valve with a Limitorque[a] motor operator was the predominant configuration. To avoid duplication, we reviewed previous applicable industry test programs. The review included the Electrical Power Research Institute (EPRI) power-operated relief valve/block valve testing at Duke Power in 1980; the Central Electric Generating Board (CEGB), United Kingdom; gate valve testing performed at Kraftwerk Union (KWU), Federal Republic of Germany; and the KWU testing performed for their own plant designs. The EPRI and CEGB work were go-no-go tests; however, the results showed that both flexwedge gate valves and parallel disc designs did have problems. The KWU work on their own gate valve designs resulted in many gate valves being replaced with globe valves. Where they could not replace the gates, they developed a structurally stiff design with rather close internal clearances. The U.S. has valves with close and large clearances, but none of our designs are as structurally stiff as the KWU designs. The KWU testing indicated that we might expect trouble based on their findings.

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a. Mention of specific products and/or manufacturers in this document implies neither endorsement or preference nor disapproval by the U.S. Government, any of its agencies, or EG&G Idaho, Inc., of the use of a specific product for any purpose.

The previous test programs did not answer our GI-87 questions, however. It must be noted here that after reviewing all of our test results, the most flexible U.S. design performed better than our moderately stiff designs. Therefore, a valve may perform correctly without being stiff. The valve design needs to be tested with the worst-case tolerances to ensure operability.

The NRC Office of Inspection and Enforcement (IE) Bulletin 85-03, "Motor Operated Valve Common Mode Failures During Plant Transients Due to Improper Switch Settings," and GL 89-10, require the utilities to develop and implement a program to ensure that the switch settings on selected safety-related MOVs are set and maintained correctly to accommodate the maximum differential pressure expected on these valves during both normal and abnormal events within the design basis. The GI-87 valves are a subset of this larger class of safety-related valves.

Industry has helped to meet these criteria by developing new MOV diagnostic test equipment and methods for in-situ valve testing. Prior to the GI-87 Valve Test Program, the motor-operator control switches settings were based primarily on standard industry practices and analysis. Very little design basis testing had been conducted in or outside of the plants. Utilities typically verified the analytically-determined MOV output torque or stem force through valve seating or backseating loads with little or no valve hydraulic loading. The GI-87 Phase I test results cast some doubts on this industry practice, primarily on whether the true design basis load can be determined analytically.

1.2 Motor-Operator Sizing

The gate valve is a high recovery positive shutoff valve and is typically used in systems where minimal pressure drop is desired when the valve is open. The design is ideally suited for isolation purposes and usually is not used for throttling flow. When the disc is in the seat, the upstream pressure load on the disc assists in sealing. This feature is less important in the flexwedge than in the parallel disc gate.

There are a number of calculations made to determine the correct operator size for a given valve application. The term sizing means that the output power available from a given motor operator is determined by its size. Typically, the larger the operator the higher the allowable output. Table 1 lists the typical maximum output in torque and stem force for the various SMB Model Limitorque operators. This table should only be used to understand what sizing means. Requalification of motor operators and other limitations on valve hardware can affect the use of a specific operator size.

The four most important calculations that are made in sizing a motor operator for a specific valve application are (a) total stem force necessary to operate the valve at its design basis load, (b) the operator torque necessary to produce that force, (c) the operator gear ratio (including the stem nut thread necessary to produce the needed valve stroke time), and (d) the size and speed of the electric motor necessary to produce the needed operator torque for that gear ratio. To be conservative, other calculations are made, such as degraded voltage concerns, which do not need to be explained for a basic understanding of motor-operator sizing. Two of these calculations, the gear ratio and the electric motor sizing, appear to be well understood and the results are repeatable in application. The required stem force and the operator torque to produce that force appear to be the areas that have not been conservatively predicted in the past for some classes of valves.

Figure 1, a cutaway drawing of a typical motor-operated gate valve, shows the components important to this discussion. The necessary forces currently defined by industry to close the valve and isolate flow must overcome the resistance imposed by three loads: (a) the disc frictional drag load caused by the differential pressure across the disc as the valve closes, (b) the stem rejection load caused by static pressure on the stem, and (c) the packing drag load. Industry has developed a set of equations for use in sizing motor operators. The first equation in this set predicts the total stem force, as detailed below. Each manufacturer modifies the variables in the equation slightly; however, in the long run the application of the equation is the same.

Table 1. Limitorque operator nuclear ratings

Model-Size	Ratio[a] Range	Rated Torque (ft-lb)	Rated Stem force (lb)	Maximum Threaded Stem Diameter (in.)
SMB-000	12.5- 30.6	90	8,000	1.375
	33.5-100.0	90		
	102.0-136.0	90		
SMB-00	9.7- 22.0	250	14,000	1.75
	23.0-109.0	250		
	114.0-183.9	190		
SMB-0	11.8- 26.1	500	24,000	2.375
	26.4- 96.2	500		
	102.6-150.8	500		
	158.3-247.0	340		
SMB-1	11.6- 25.7	850	45,000	2.875
	27.2- 88.4	850		
	92.4-171.6	780		
	191.7-234.0	625		
SMB-2	10.6- 25.6	1800	70,000	3.5
	26.2- 82.5	1800		
	84.8-150.0	1250		
	153.0-212.5	950		

a. Unit overall gear ratio.

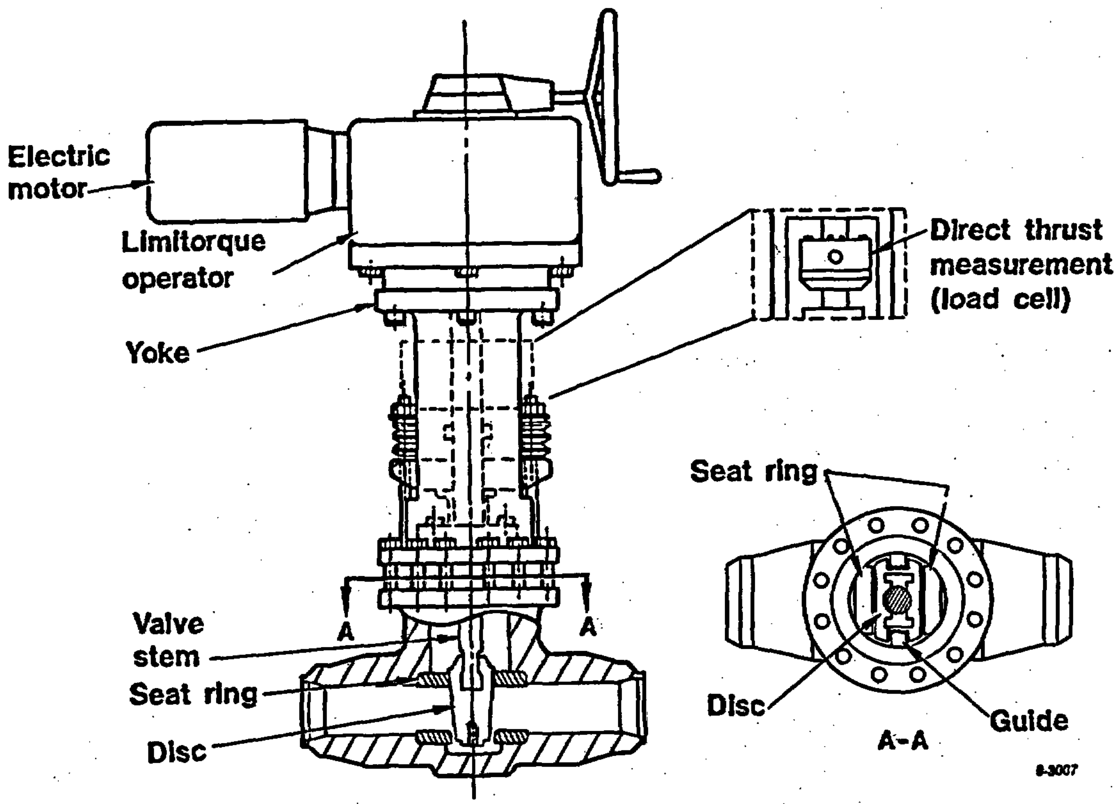

Figure 1. Typical motor-operated valve showing components important to calculating stem force. Two of the test valves were modified by installing a load cell in the valve stem.

$$F_t = \mu_d A_d \Delta P \pm A_s P + F_p \quad (1)$$

where

F_t = total stem force

μ_d = disc factor
A_d = disc area
ΔP = differential pressure
} dynamic component

A_s = stem cross-section area
P = stem pressure
F_p = packing drag load (a constant).
} static component

For wedge-type gate valves, the disc factor (μ_d) normally used by the industry in Equation (1) is 0.3. Note that in this equation the stem rejection load can be either positive or negative, dependingon whether the valve is closing or opening. This is because the stem rejection load is always in a direction out of the valve body; this load resists valve closure and assists in opening the valve. The packing load is typically constant and depends on the packing design, the gland nut torque, and the direction of operation.

The equation is divided into two components, which will be referred to in the analysis found later in this report. The components are (a) the dynamic component, which includes the disc load due to differential pressure, and (b) the static component, which is the sum of the stem rejection and packing drag loads. The pressure values (P and ΔP) used in the force equation are supplied to the valve manufacturer by each individual utility. Determining the motor-operator torque necessary to produce that force is complicated by the fact that motor operators control output torque, not stem force. Thus, in determining the necessary output torque one must consider the conversion of operator torque to stem force. The torque-to-stem force relationship normally used in sizing motor operators depends on a stem factor calculation given by

$$T = \mu_s F_t \quad (2)$$

where

T = operator torque

μ_s = stem factor

F_t = total stem force [from Equation (1)].

The stem factor used in Equation (2) is a function of stem diameter, thread pitch and lead, and the coefficient of friction between the operator stem nut and the valve stem. As in Equation (1), the only variable that cannot be measured in the stem factor equation is the coefficient of friction. Normally, it is assumed that only damage and lubrication of the stem/stem nut threads can significantly alter the stem coefficient of friction. Limitorque personnel, in their diagnostic work, have measured coefficients of friction from 0.10 to 0.20 in actual operation. Losses internal to the motor operator, up to the capacity of the electric motor, will typically be accounted for by the torque spring/switch position. Losses in the stem factor will not be accounted for by the motor operator.

The problem with the conversion of torque-to-stem force (stem factor) is not in conservatively bounding it in the sizing calculation. It is that the stem factor appears to change with stem load. This will complicate utility efforts to comply with regulatory recommendations to develop and implement a program that will ensure that safety-related motor-operator torque switch settings (the switch that regulates the motor-operator output torque) are chosen, set, and maintained correctly to accommodate the maximum differential pressure load expected on the valves during both normal and abnormal events within the design basis. Additionally, MOV torque-to-force relationships can vary with age and maintenance. Torque springs age, changing the torque switch setting in comparison to output torque. The stem factor can change for two primary reasons: (a) lubrication quality at the valve stem-to-motor-operator stem nut interface, and (b) degradation of the threads of either component. The stem factor can also improve with wear between the stem and stem nut threads.

The INEL believes the biggest motor-operator sizing problem today is verifying the capability of the operators in the plants. This problem is compounded for gate valves and to some degree for all rising stem motor-operated valves according to NRC GI-87 valve test results. This indicates that the variables used by industry in the past for determining valve opening and disc force [Equation (1)] at high flow were not conservative, and that the stem factor may vary with load, making it very difficult for the utility to diagnostically determine operator capability in place without design basis testing. This is not always possible in a plant.

Motor-operator sizing topics that apply to both GI-87 and GL 89-10 will be discussed in more detail in later sections of this report.

2. APPROACH TO TESTING

Six full-scale, representative nuclear valve assemblies were tested under various normal operations and design basis pipe break conditions for the RWCU, HPCI, and RCIC systems. Table 2 lists the test valves and motor operators used in both Phase I and II test programs.

As shown in Table 2, the two valves used in Phase I were reused in Phase II. Phase I Valve A was refurbished and became Valve 1 in Phase II. The valve's internal manufacturing tolerances allowed the disc to tip downstream during the Phase I testing, causing damage to the disc guide surfaces. Gate valves are bidirectional; therefore, the valve was turned around for Phase II, reversing flow through the valve to see how the internal manufacturing tolerance stackup in the other direction affected valve performance. Valve B from Phase I was returned to its manufacturer for a valve disc replacement and became Valve 2 in Phase II. The valve disc or gate in Phase I was equipped with hardfaced disc guides, representative of valve assemblies built after 1970. For Phase II, the hardfaced guide disc was replaced with a normal material disc guide, representative of those valves made before 1970. With the exception of Valve B in Phase I, all other valves in both phases had normal carbon steel disc guide surfaces representative of the largest majority of the installed gate valves. All valves in both phases had hardfaced sealing surfaces on both the body and the discs. Valves 3, 4, and 5 were new valves obtained through canceled nuclear plant surplus. These three valves and the two valves used in Phase I were returned either to their manufacturer or to a nuclear valve service center for refurbishment and/or inspection prior to their use in Phase II testing. Valve 6 was manufactured new for the Phase II Program. Prior to Phase II testing, each valve body was assembled with instrumented spool pieces and instrument taps as shown in Figure 2.

Table 2 also identifies the 460-Vac, 3-phase, 60-Hz Limitorque electric motor operators used on the valve assemblies. For Phase II, the same SMB-O-25 was used on all three 6-in. valve assemblies, and the same SMB-1-60 was used on the three 10-in. valve assemblies. The motor-operator stem nuts and helical reduction gears were changed to accommodate the valve stem thread pitch and lead and to establish, as close to as possible, a 30-s stroke time for all six valves. For valve stem force measurements, two of the valves were instrumented with direct stem-mounted load cells, as shown in Figure 1. The other valves used a set of four load cells mounted between the valve yoke and motor operator. During final checkout of the motor operators, Limitorque ran a special torque spring deflection versus operator output torque calibration on their dynamometer. By using a linear variable displacement transformer (LVDT) and the Limitorque torque spring deflection versus operator output torque relationships, we were able to monitor apparent motor-operator torque on-line during the test program. Figure 3 shows the general location of the valve response instrumentation used during the Phase II test program.

The Phase II test program was performed in the late summer and early fall of 1989 at the KWU facilities near Frankfurt, West Germany. Two test loops were used. Figure 4 shows the loop used for the 6-in. valve tests; Figure 5 shows the loop used for the 10-in. valve tests. Both test loops used a 22-MW oil-fired boiler for heating and test media propellant. The 6-in. valve test stand also had the capability of being charged with gaseous nitrogen for cold water high flow testing. Both test stands were equipped with bypass lines around the loop blowdown device. These lines were used to establish normal flow through the valves for normal-service functional testing. Maximum flow was established on the 6-in. valve test loop through a rupture disc and on the 10-in. valve loop through quick opening valves.

The data collection objectives for Phase II testing were driven by the lessons learned from the Phase I program, where the performance of the valves under higher loadings was unlike previous industry claims for valve performance. Valve thermal hydraulic inlet conditions appeared to influence these performance problems (i.e., the measurements made during the Phase I program

Table 2. Valve and operator information

Test Hardware

Valve Manufacturer and Valve Identification

	Valve Designation				
	Phase I	Phase II	Size	Class	Type
Anchor/Darling[a]	A	1	6-in.	900 lb	Flexwedge Gate
Velan[a]	B	2	6-in.	900 lb	"
Walworth		3	6-in.	600 lb	"
Anchor/Darling		4	10-in.	900 lb	"
Wm. Powell		5	10-in.	900 lb	"
Velan		6	10-in.	600 lb	"

Motor-Operator Manufacturer	Size	Valve Used
Limitorque	SMB-0-25	1, 2, 3 and B
Limitorque	SMB-1-60	4, 5, and 6
Limitorque	SMB-2-40	A

a. Valves A and B from Phase I were refurbished for Phase II as Valves 1 and 2.

were not sufficient to allow a thorough analysis of specific thermal hydraulic influences). The Phase II measurements, instruments, and data-collection tools were optimized as thoroughly as practical to characterize the test valves' and motor-operators' performance at the various test conditions. Inputs to these measurement and data collection schemes included suggestions from experts at the INEL, from the NRC staff at planning meetings, and from industry experts at several review meetings.

To quantify the test loop thermal hydraulic conditions and valve response, an average of 70 channels of information were measured and recorded on a high-speed tape recorder. The tape recorder was an integral part of the INEL data acquisition system (DAS). The DAS transducer excitation voltage was measured at the transducer and the system adjusted the internal voltage to account for line losses. The outputs from the transducers were then measured by the DAS and committed to the tape.

The DAS was also configured to make incremental calculations of the stem force throughout the closing and opening cycles using the measured values for the parameters in Equation (1) (differential pressure, disc area, etc.) and a constant disc factor, either the common 0.3 or a more conservative 0.5. We performed our calculations in this manner to see how well the equation modeled the actual stem forces during the entire cycle. A comparison of the calculated forces to the measured forces shows where the deviations

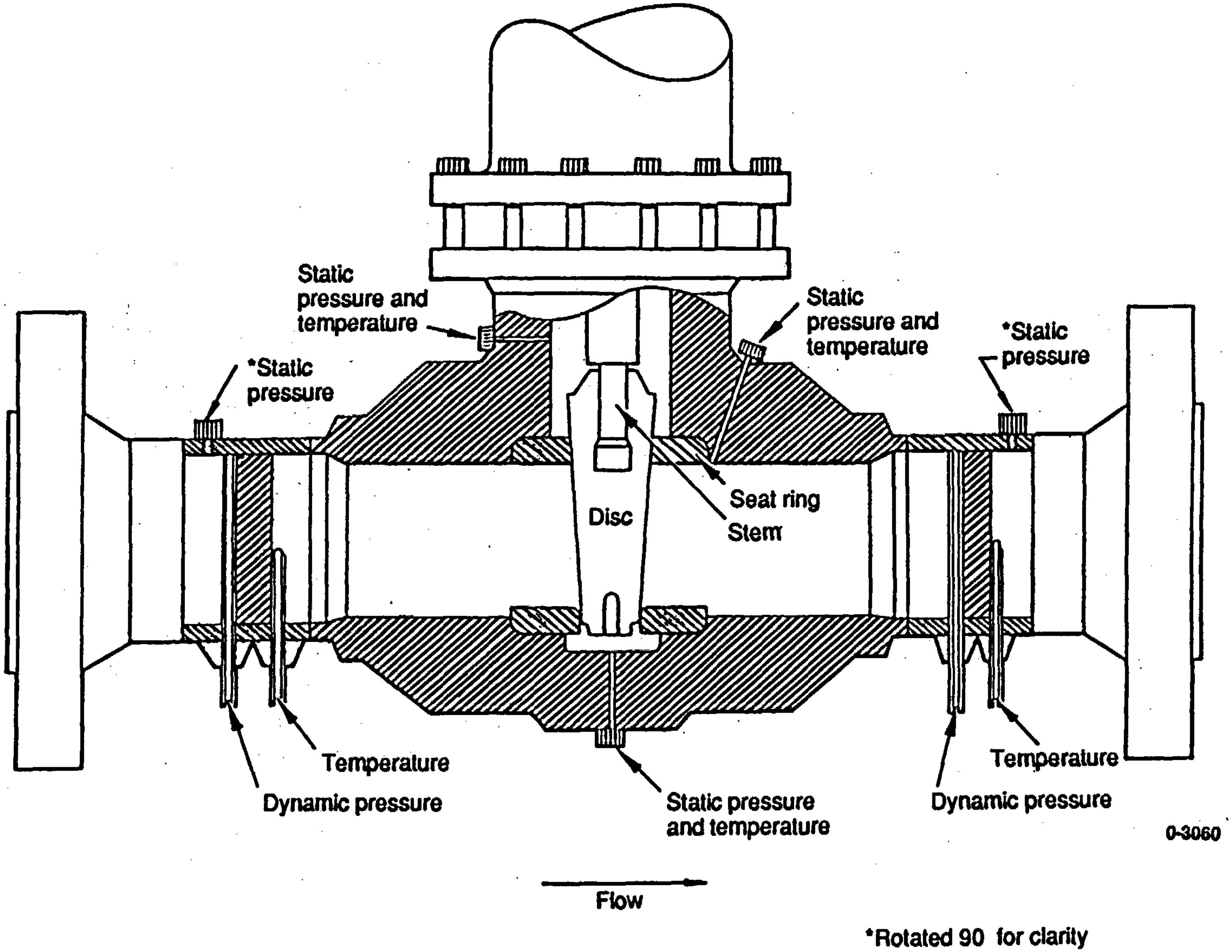

Figure 2. Valve assembly showing instrumented spool pieces.

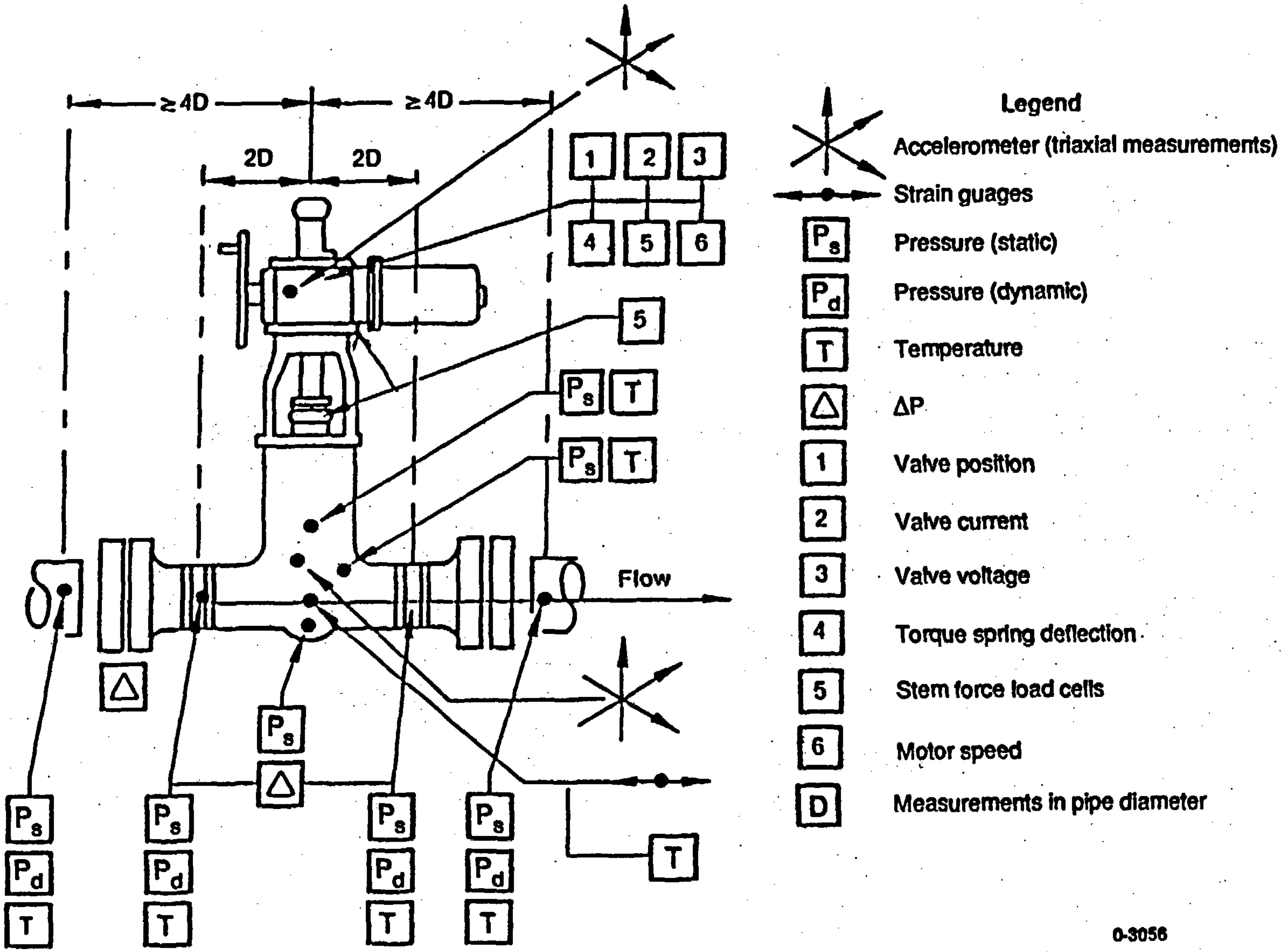

Figure 3. Phase II valve assembly and local piping instrumentation.

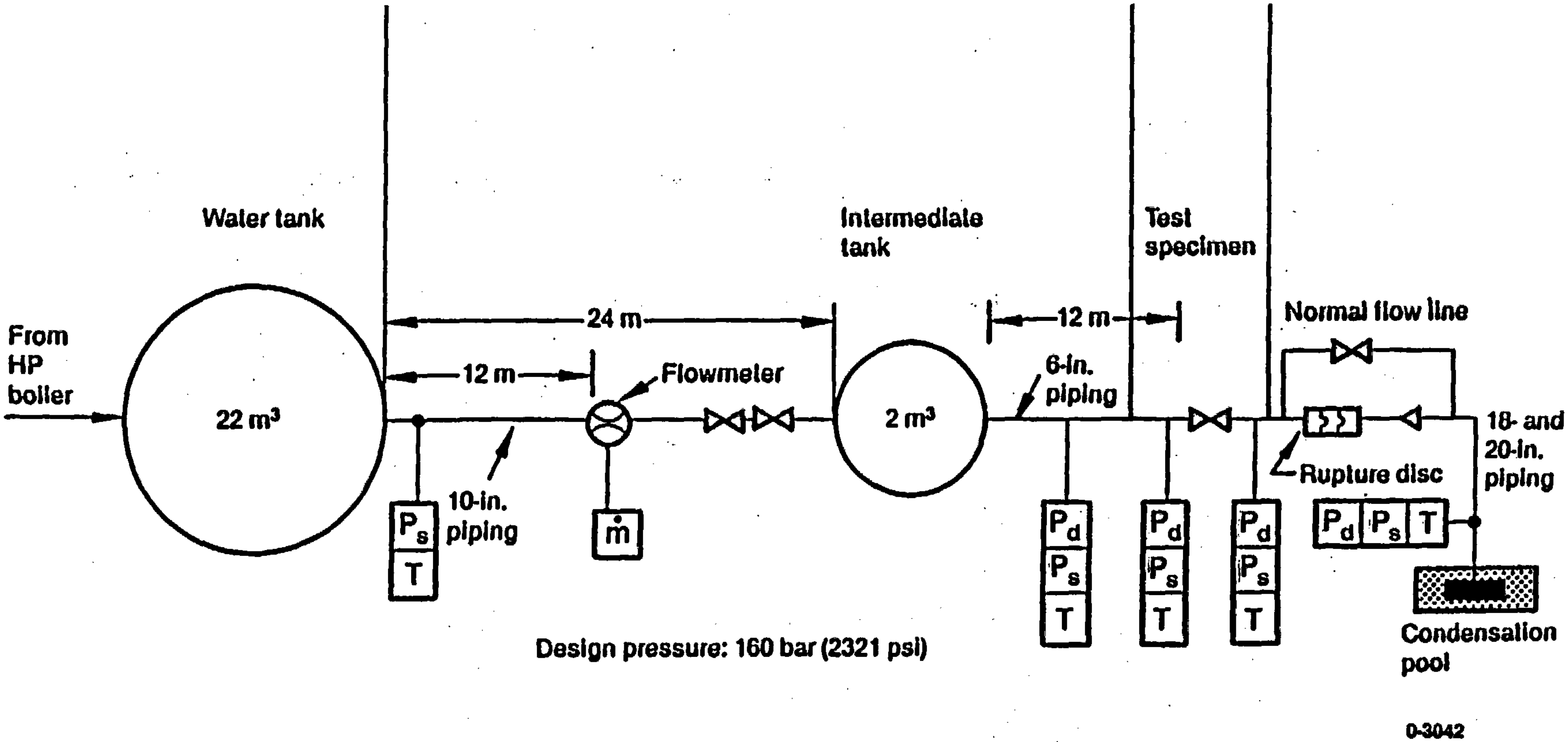

Figure 4. Simplified KWU 6-in. valve test loop showing configuration, lengths, and piping sizes.

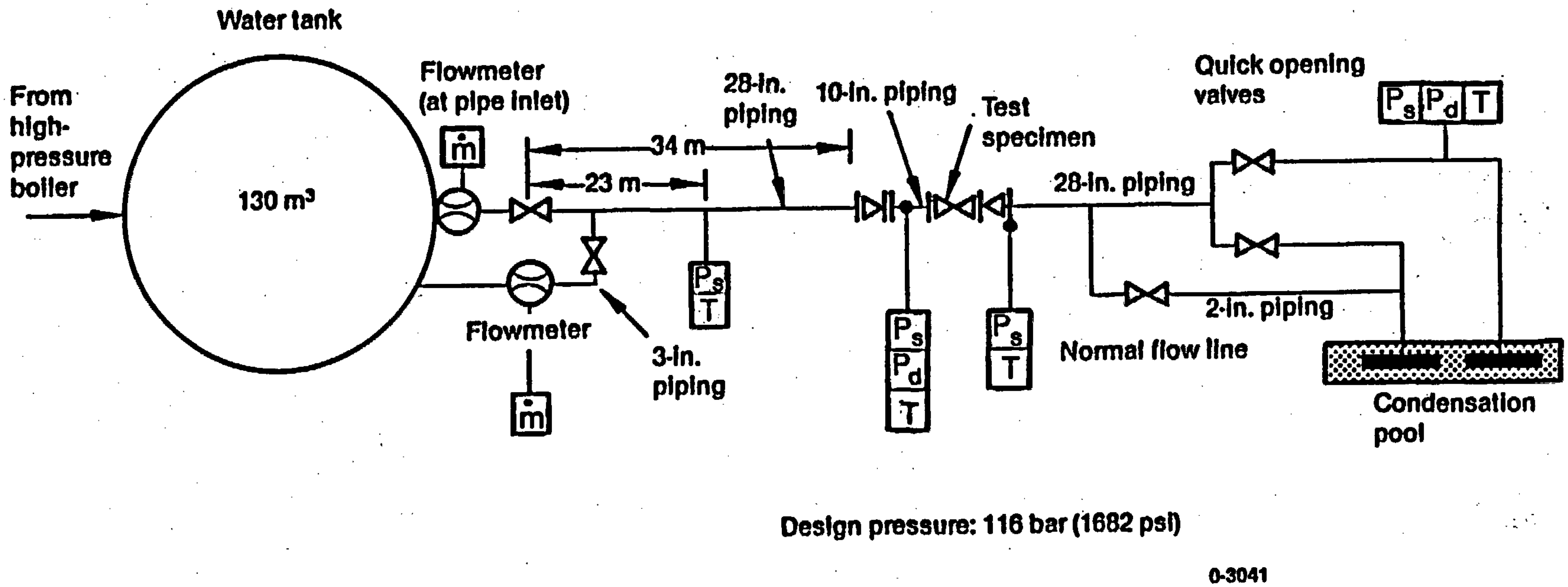

Figure 5. Simplified KWU 10-in. valve test loop showing configuration, lengths, and pipe sizes.

start to take place and allows us to look at the fluid conditions and other parameters and determine if influences other than the terms in Equation (1) affect the stem forces during valve operation.

At various times during the test programs, parallel diagnostic measurements were made by Bechtel-KWU Alliance, General Physics, Liberty Technology, Limitorque, Movats, Westinghouse, and Wyle Laboratories. This allowed the diagnostic vendors to compare measured loadings, similar to those that might be developed during in-plant testing, to design basis loadings, which typically cannot be developed in the plant.

3. TESTING

Six qualification tests and seventeen flow interruption tests were performed. Seven of the flow interruption tests were design basis tests; the other ten were parametric studies. The parametric studies were an attempt to understand the effects of pressure, temperature, and fluid properties on valve performance. The fluid conditions and valve operating responses provided information concerning valve and motor-operator performance at various valve loadings.

The two lists below summarize the qualification and flow interruption test procedures for each valve. The basic test procedure for each valve through the design basis flow interruption test was nearly the same, with a few exceptions (listed later).

The following list is an overview of the test procedure performed in accordance with the test plan for each valve through the design basis flow interruption test, which was always performed first on each valve assembly:

- The valve was installed in the test loop
- The motor operator was installed and limit control switches were set
- Instrumentation was installed
- The torque switch was set
- A baseline opening and closing test without pressure was performed
- Cold leakage test (Annex A of ANSI/ASME B16.41)
- Cold cyclic test (Annex B of ANSI/ASME B16.41)
- Hot cyclic test (Annex C of ANSI/ASME B16.41)
- Flow interruption test at the normal operating pressure and temperature (NOP/NOT) for the valve's representative system (Annex G of ANSI/ASME B16.41). The details of this step are provided below.

Exceptions to this basic procedure are as follows:

- Because of operator limitations at the very high torque switch settings used, some of the under-voltage tests recommended by the standard in hot and cold cycling were not performed.
- The torque switch settings in some cases were increased after hot cycling to ensure valve closure. (Our objective was to close the valve and determine from the measurement how much stem force was actually required to isolate flow and seat the valve. In one case (Valve 1), we underestimated the closing force; we obtained flow isolation but did not fully seat the valve.)

The flow interruption test consisted of the following numbered steps. The numbers reference the headers of the Phase II plots used in this report:

- Bring the test loop and valve up to NOP/NOT (Step 7)
- Close the valve at NOP/NOT without flow (Step 13)
- Depressurize the downstream side of the valve (Step 15)
- Open the valve against NOP/NOT upstream pressure (with the downstream valves closed) (Step 17)
- Establish normal flow through the test valve by opening the valve in the normal flow line (see Figures 4 and 5)
- Close and reopen the test valve at normal system flow (Step 18)
- Close the valve in the normal flow line

- Establish line break flow through the test valve and close the test valve (Step 25)

- Reopen the test valve 30% open and reclose at maximum flow (Step 26)

- Depressurize the test loop and open and close the test valve while still at temperature without system pressure (Step 30).

Following the flow interruption tests, we reviewed the quick look plots, which served two purposes:

- To determine if the valve had been damaged during the test. Valve stem force plots are very good indicators of valves sustaining damage during the high flow tests (see Figure 6).

- To determine if the thermal hydraulic conditions at the valve inlet met the test objectives.

If the valve was damaged by the loading (as determined from a review of the stem force plot), we stopped testing and installed the next valve. If the valve was not severely damaged and the thermal hydraulic conditions were met, we then subjected the valve to parametric studies (varying the pressure and/or temperature). If the thermal hydraulic conditions were not met, we repeated the test.

Table 3 provides the test matrix for the flow interruption test sequences performed on each valve. The temperatures and pressures listed for each sequence include the target pressures and temperatures for a nearly closed valve. Facility capability limitations necessitated that some tests be started with a slightly higher pressure than the target pressure. In some cases, the valve was started from less than 100% open in the maximum flow closures so that the target pressure would be achieved as closely as possible when the valve was nearly closed.

Table 4 provides the maximum working pressures for the valve classes tested in this program and the test pressures for the leakage and cyclic tests. Test pressures were not always maximums for the valve class because of facility limitations; however, they did bound the qualification pressure for the intended service.

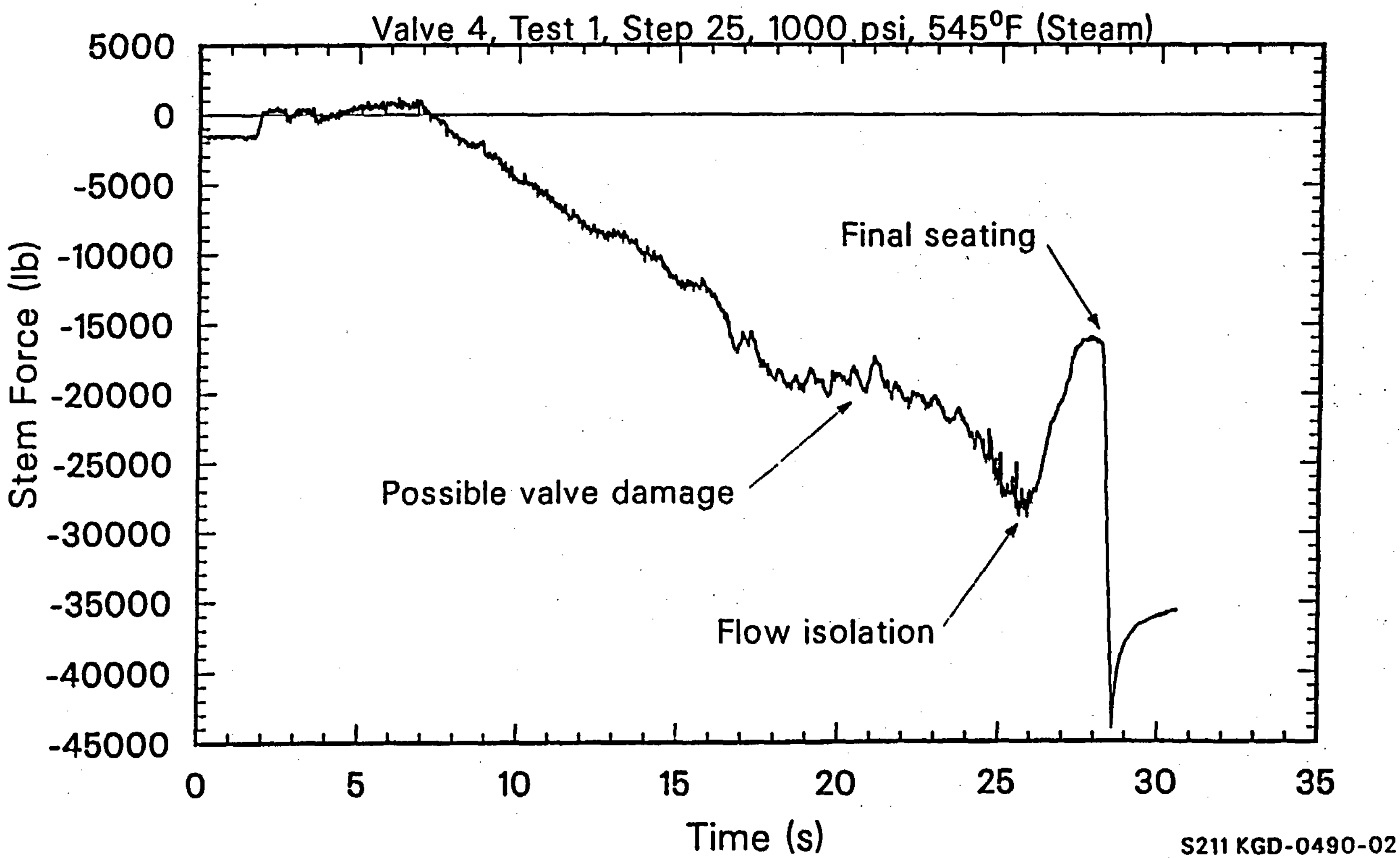

Figure 6. Valve stem force history from Phase II testing shows nonlinear performance of the valve, indicating valve damage.

Table 3. Flow interruption test target temperatures and pressures

Valve No.	Test No.	Target Pressure (psig)	Actual Pressure (psig)	Target Temperature (°F)	Actual Temperature (°F)	Media
6-in. Valve Tests						
1	1	1000	900	530	520	Hot water
2	1	1000	950	530	520	Hot water
2	2	1000	1040	545	550	Steam
2	3	1000	750	<100	<100	Cold water
2	6A	600	600	300	450	Hot water
2	6B	1000	1000	430	470	Hot water
2	6C	1400	1300	480	520	Hot water
3	1	1000	920	530	520	Hot water
3	5	1200	1100	550	550	Hot water
3	7	1400	1300	580	570	Hot water
10-in. Valve Tests						
4	1	1000	750	545	510	Steam
5	1A	1000	800	545	520	Steam
5	1B	1400	1040	590	550	Steam
6	1A	1000	990	545	580	Steam
6	1B	1400	1400	590	590	Steam
6	1C	1200	1100	570	550	Steam

Table 4. Valve qualification test pressures

		Maximum ANSI working pressure by pressure class for these alloys	
Valve No.	Class	Temperature (°F)	Maximum Working Pressure (psig)
1, 2, 4, 5	900 lb	<100	2250
	900 lb	600	1815
3, 6	600 lb	<100	1500
	600 lb	600	1210

Cold Leakage Test Annex A

Valve No.	Temperature (°F)	Test Pressure (psig)
1	<100	2200
2	<100	2200
3	<100	1500
4	<100	2200
5	<100	2200
6	<100	1500

Cold Cyclic Test Annex B

Valve No.	Temperature (°F)	Test Pressure (psig)
1	<100	1650
2	<100	1650
3	<100	1200
4	<100	1600
5	<100	1600
6	<100	1200

Hot Cyclic Test Annex C

Valve No.	Temperature (°F)	Test Pressure (psig)
1	600	1650
2	600	1650
3	550	1200
4	610	1600
5	610	1600
6	550	1200

4. TEST RESULTS AND INTERPRETATIONS

4.1 Generic Issue 87

GI-87 addresses whether containment isolation valves will close in the event of a pipe break outside of containment and downstream of the isolation valves. (The question also applies to the RCIC steamline and the RWCU supply line.)

Test results provided positive evidence that, given enough stem force, the flexwedge gate valve can close at pipe break flows. Test results also showed that the stem force necessary to close each of the test valves was considerably higher than the force that would normally be predicted using the standard industry motor-operator sizing equation. Thus, the potential exists for undersized operators on these containment isolation valves.

Because of our Phase I test experience, where we tested two 6-in. flexwedge gate valves at RWCU linebreak flows, we expected higher stem force requirements for closing the Phase II valves. Therefore, we sized the operators and set the torque switches to produce stem forces high enough to close the valves under any credible loading based on our previous experience. Our plan was to close the valves and determine from the measurements how much stem force was actually required to isolate flow and seat the valves.

4.2 Flexwedge Gate Valve Stem Force Requirements

Figures 7 through 9 show the stem force histories for the three 6-in. valves closing against high energy hot water that were representative of line break flow in the RWCU supply line. Figures 10 through 12 show the stem force histories for the three 10-in. valves closing against high energy steam flow representative of line break flow in the HPCI turbine steam supply line, and Figure 13 shows the stem force history for one of the 6-in. valves closing against high energy steam flow representative of linebreak flow in the RCIC turbine steam supply line.

The stem force histories in these figures were compared with the forces that were calculated using Equation (1) with a constant disc factor at the commonly used industry 0.3 and at a more conservative 0.5. This was done, as previously explained, with the DAS using real time values for the variables in the equation. The actual port size for each valve (as provided by the manufacturer) was used in the area calculations instead of other mean diameters, which are sometimes used by the industry. The ratio of the actual port size-to-pipe diameter varies with the class of valve and manufacturer, resulting in a variation in the degree of conservatism. All conservatisms were removed from the calculations so we could compare the actual stem force histories with the calculated stem force histories based on Equation (1).

The stem force histories are plotted against time, representing the stroke time of the valve for that closing. Due to facility limitations, some tests were not started with the valve in the fully open position. During the closing stroke of the valve, the stem is in compression; thus, the negative convention of the stem force history. The more negative the stem force history the greater the required stem force.

The test results indicate there are generally two types of valve stem force responses, those bounded by the calculations and those that are not. Those that are bounded we call predictable and those that are not we call nonpredictable. Figures 7 through 13 show several different shapes for the stem force histories. As we stated, the stem force calculations are made through the DAS and the same real time forces are acting on the calculations as on the valve disc. The only difference between the stem force calculations and the measured stem force history is that the calculations have an assumed disc factor [Equation (1)] and they are not influenced by fluid properties. Fluid property effects will be discussed later in the report; however, the differences for the RWCU 10•F subcooled hot water and the HPCI-RCIC steam conditions are minor. As we examine Figures 7 through 13, the major difference in the relationship between actual and calculated stem forces is due to the pressure distribution through the valve and the disc guiding and sealing surface drag, which are both lumped in to the disc factor term in Equation (1). None of the actual stem

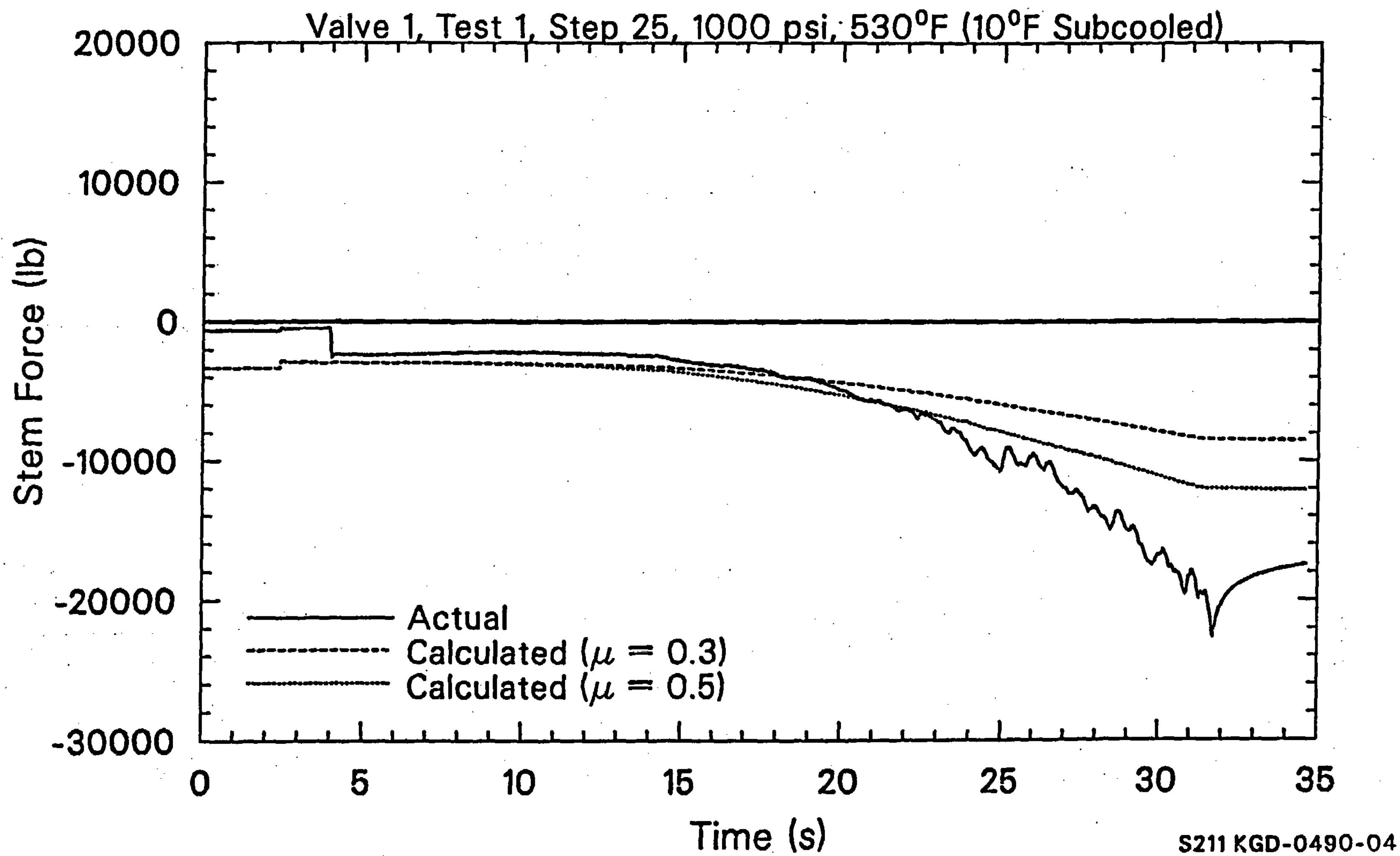

Figure 7. Valve 1 in Phase II, 6-in. RWCU system valve, same as Valve A in Phase I, nonhardfaced guides, refurbished, and tested with the opposite end as the inlet at line break flow, comparing actual versus predicted stem force.

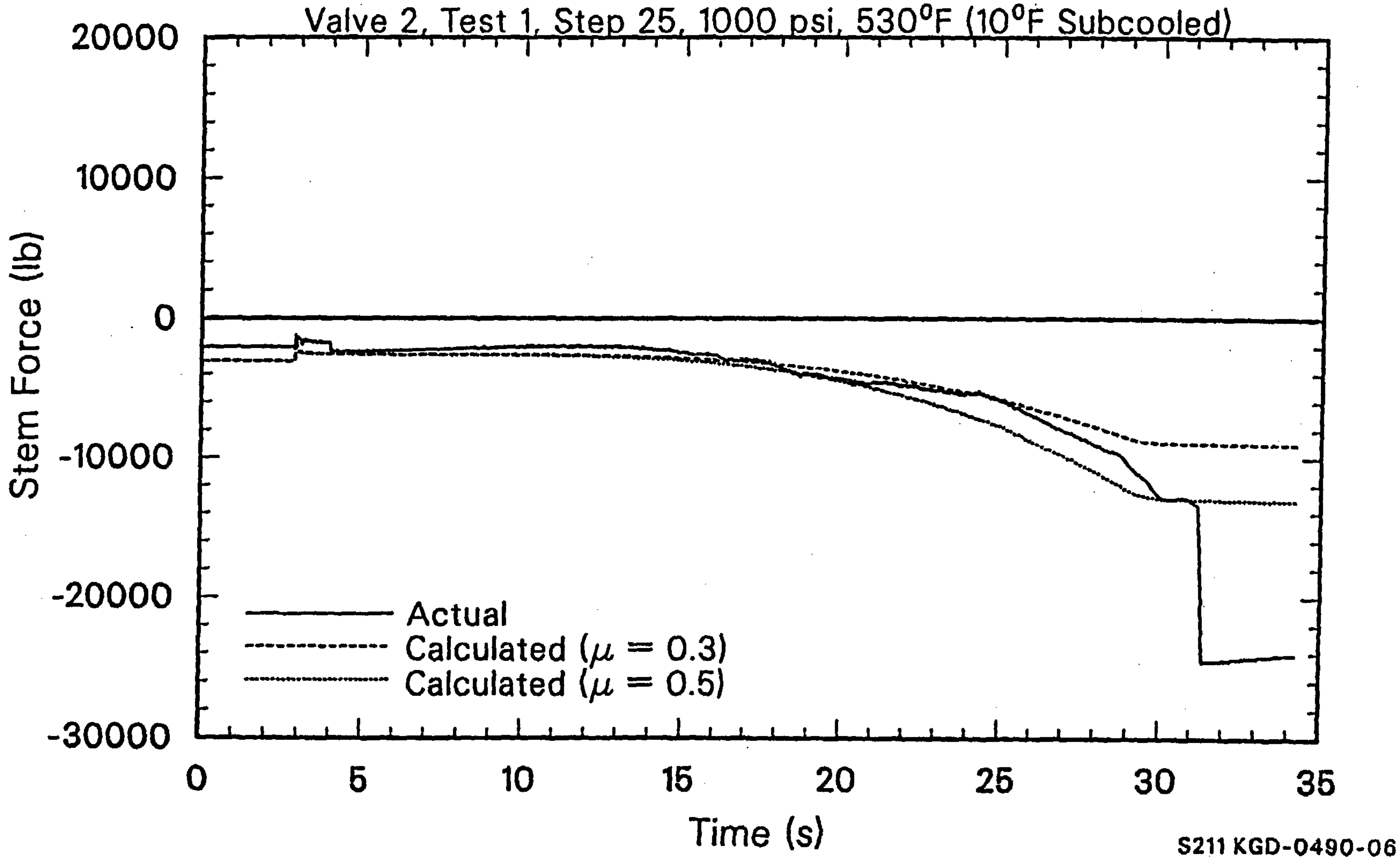

Figure 8. Valve 2 in Phase II, 6-in. RWCU system valve, same as Valve B in Phase I except nonhardfaced disc guides, line break flow, comparing actual versus predicted stem force.

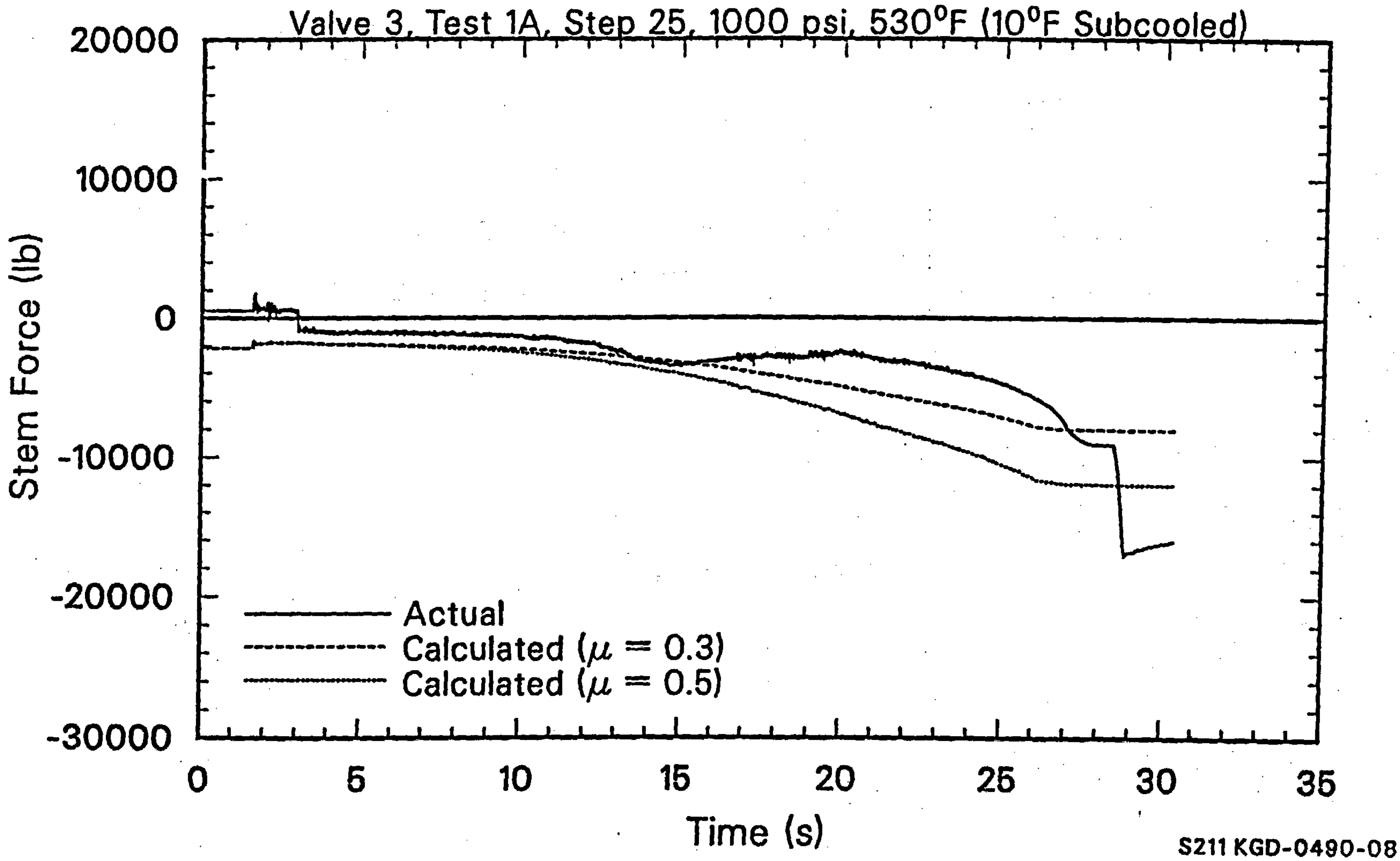

Figure 9. Valve 3 in Phase II, 6-in. RWCU system valve, nonhardfaced guide surfaces, line break flow, comparing actual versus predicted stem force.

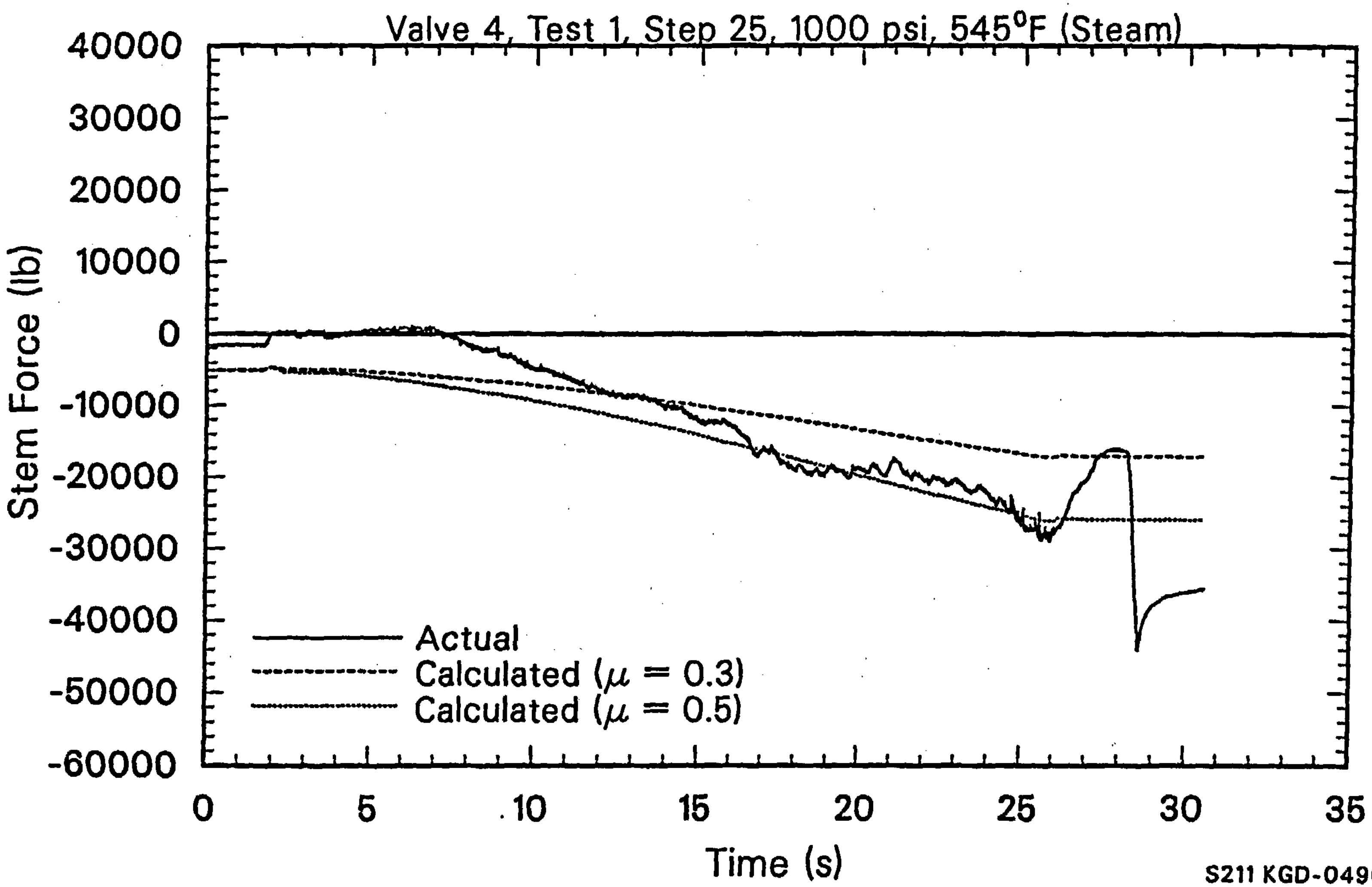

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Figure 10. Valve 4 in Phase II, 10-in. HPCI system valve, nonhardfaced guide surfaces, line break flow, comparing actual versus predicted stem force.

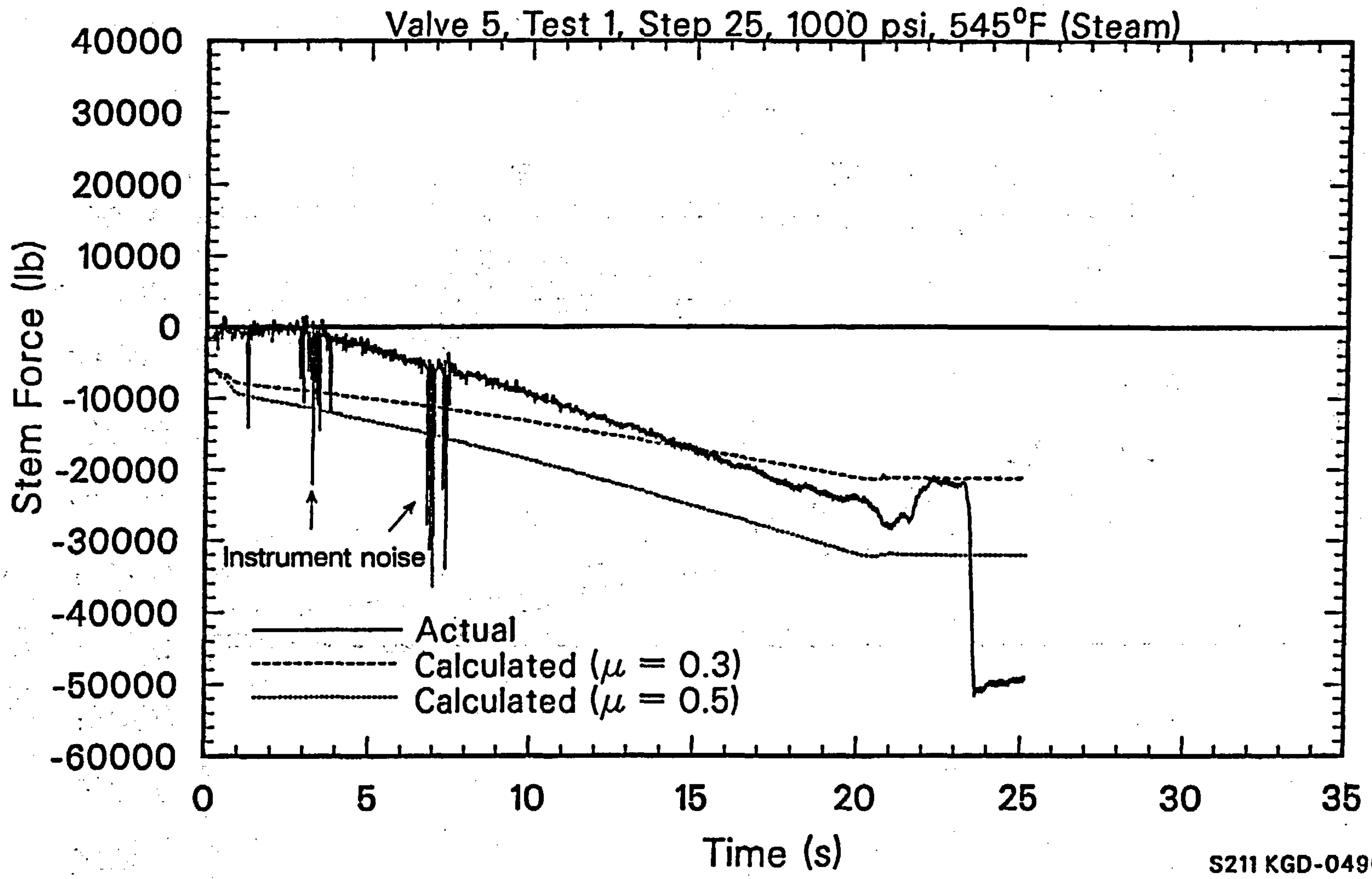

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Figure 11. Valve 5 in Phase II, 10-in. HPCI system valve, nonhardfaced guide surfaces, line break flow, comparing actual versus predicted stem force.

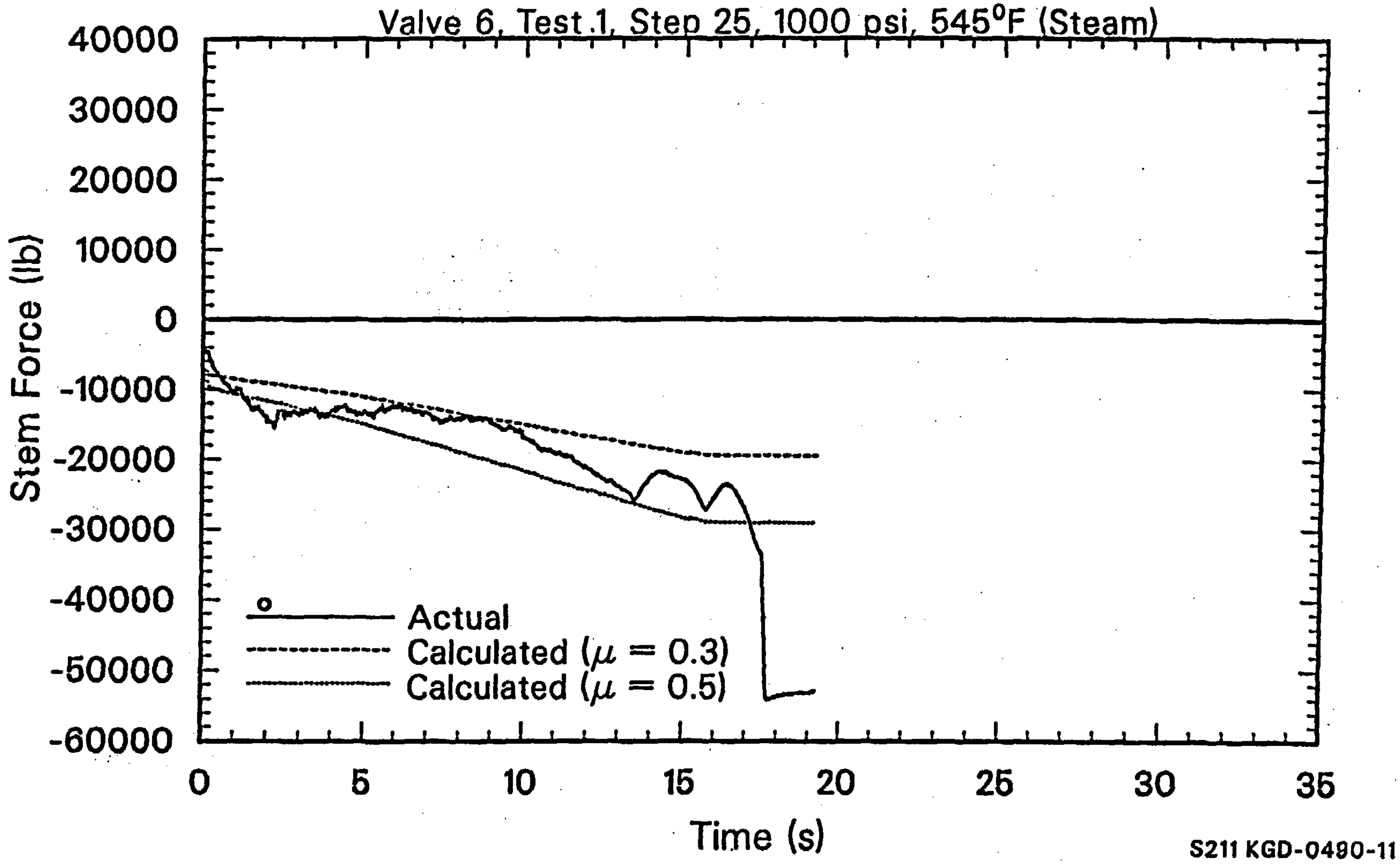

Figure 12. Valve 6 in Phase II, 10-in. HPCI system valve, nonhardfaced guide surfaces, line break flow, comparing actual versus predicted stem force.

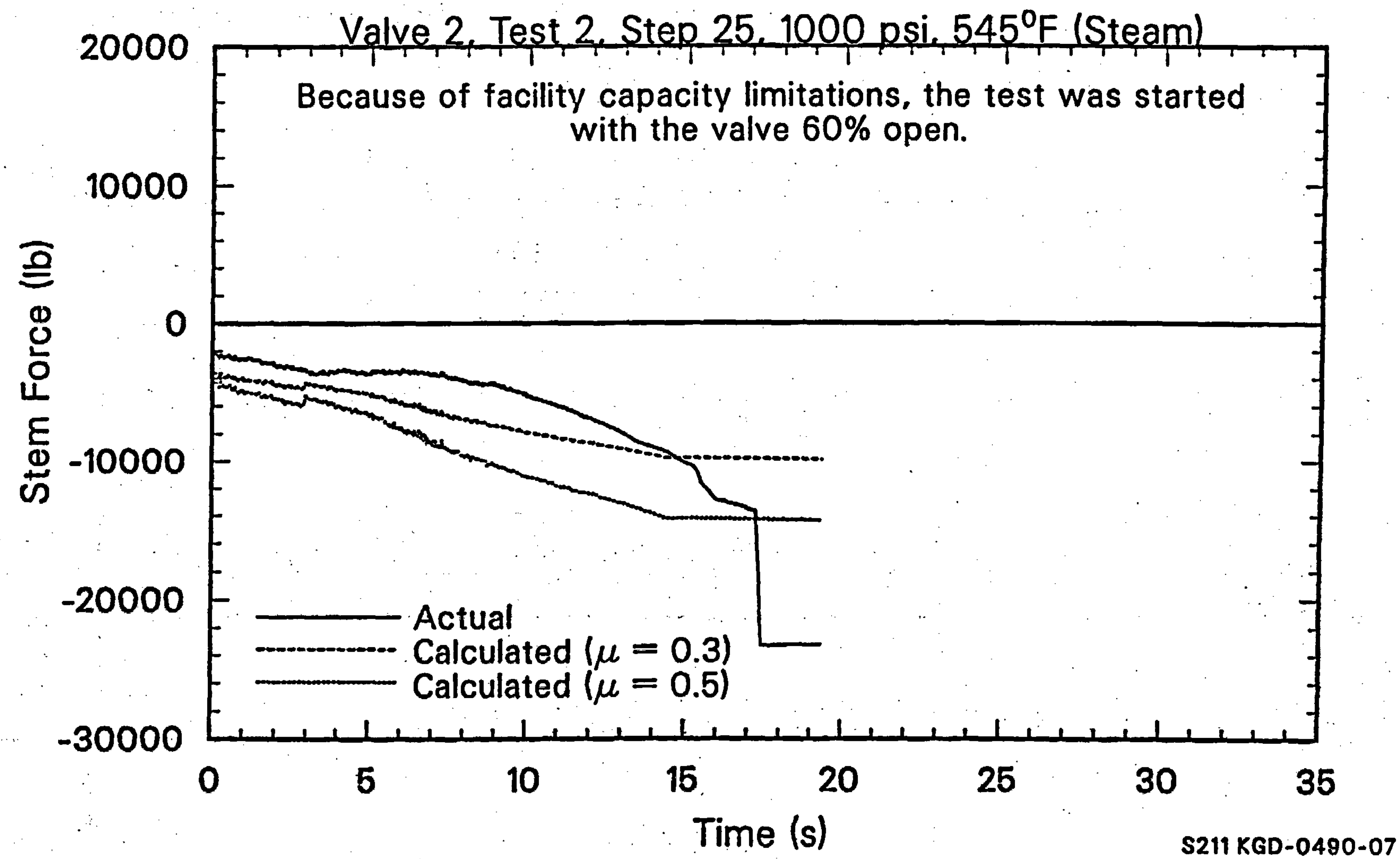

Figure 13. Valve 2 in Phase II, 6-in. RCIC system valve test, closing on line break steam flow, comparing actual versus predicted stem force.

force histories exhibit linear behavior; however, Valve 2 (Figures 8 and 13, tested with hot water and steam, respectively) follows the general trend of the calculations with a slowly increasing load. Thus, the actual valve response is not modeled by Equation (1) calculated stem force histories, but it can be bounded with a conservative disc factor. This response we have called predictable.

The nonpredictable valve response is exhibited in Figures 7, 10, 11, and to some degree, in Figure 12. The stem force history in Figure 7 departs radically from the calculations, and the stem force load recovers at seating in Figures 10 and 11. There is no recovery in Figure 7 because the valve did not fully seat. We will be discussing this anomaly later in the report; however, in the Phase I test series we did seat the valve and it exhibited the same characteristics as seen in Figures 10 and 11. You will also note commonality (the sawtooth characteristic) in the three stem force histories. Posttest inspection of these valves showed this jagged response to be representative of internal damage occurring to the valve guide and seating surface. The damage took three forms: plastic deformation of the guides, ripping and gouging of the guide surfaces, and the disc machining off part of the body seat surface during the closing stroke. The INEL believes all three of the damage mechanisms were allowed by the clearance between the disc and valve body guide surfaces. The clearances were great enough to allow the flow forces to tip the disc downstream, as demonstrated in Figure 14. Damage was observed when the clearances were great enough to allow the nose of the disc to contact the seat sealing surface. The tipping disc also reduced the contact area on the guide surface, concentrating the disc load on a smaller surface area. The yield strength of the material was exceeded, resulting in one of the above damage mechanisms depending on the relative roughness of the disc body guide surfaces. The rougher surfaces ripped and gouged the smoother surfaces. Some valves exhibited combinations of both guide and sealing surface damage. Those valves without antirotation features on the stem also showed that stem torque reacting on the disc influenced the damage. One side of the disc was damaged more than the other. Figures 15 through 19 are photographs of representative damage.

These results indicate that for the valves we have called nonpredictable, the disc factor (coefficient of friction) is not a product of sliding friction or other internal valve loadings, but rather the variable of additional stem force necessary to overcome internal valve damage. The force necessary to overcome internal damage is valve-specific depending on the internal clearances. To determine this additional force, one must test each valve. The test would damage the valve, requiring refurbishment. Refurbishment will change the internal clearances, negating the results of the test. This Catch-22 may remain until we improve the internal design of these valves.

Valve 3 (Figure 9) falls into the predictable category, but from a different response profile than the others. All of the other valves are built with rigid body guides (i.e., there is no flexing in the disc body guide interface). Valve 3 has a relatively flexible, removable guide. It is anchored only at the bottom of the body and at the top of the bonnet; therefore, the guide is free to flex the entire length of the body (Figure 20 shows the construction details). Figure 9 shows a dip in the force history at about the 15-s time line. This plot detail and the posttest inspection of the guide, disc, and body seat show the guide was plastically deformed by the large flow load acting on the disc, allowing the disc to make full seat contact during closing. The guide was not stiff and permitted the disc to translate in the direction of flow. The disc closed, bearing squarely on the hardfaced seat. The sealing surfaces on the disc are also hardfaced. This hardface-to-hardface contact resulted in the lowest apparent disc factor (compared to calculation) of all of the valves tested. All other guide designs were rigid and resisted the flow loads. Because of this flexibility, this guide design conformed to the load and the damage was not as great as that of some other designs. The disc-on-seat loading did cause some seat damage, however, and the valve leaked after the first blowdown loading.

From the stem force history and posttest inspection, it appeared that Valve 6 (Figure 12) suffered a combination of both disc guide surface damage and seat damage. These did not occur simultaneously, however. During the period of 0-5 seconds, it appeared that damage was occurring to the guide. Then, as more of the disc entered the flow stream and more seat contact was made, the disc straightened out.

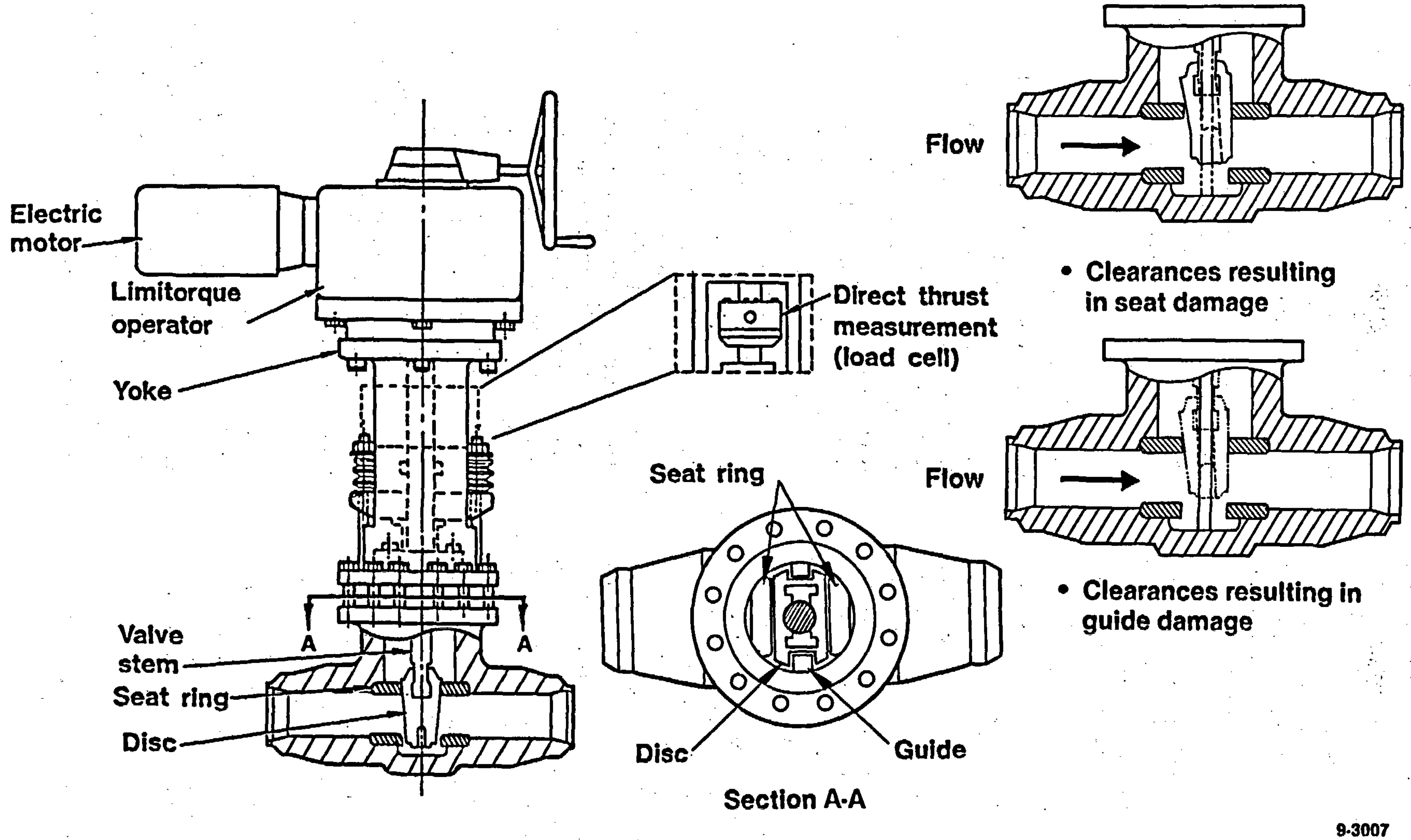

Figure 14. Gate valve cross section showing possible valve tippage during flow.

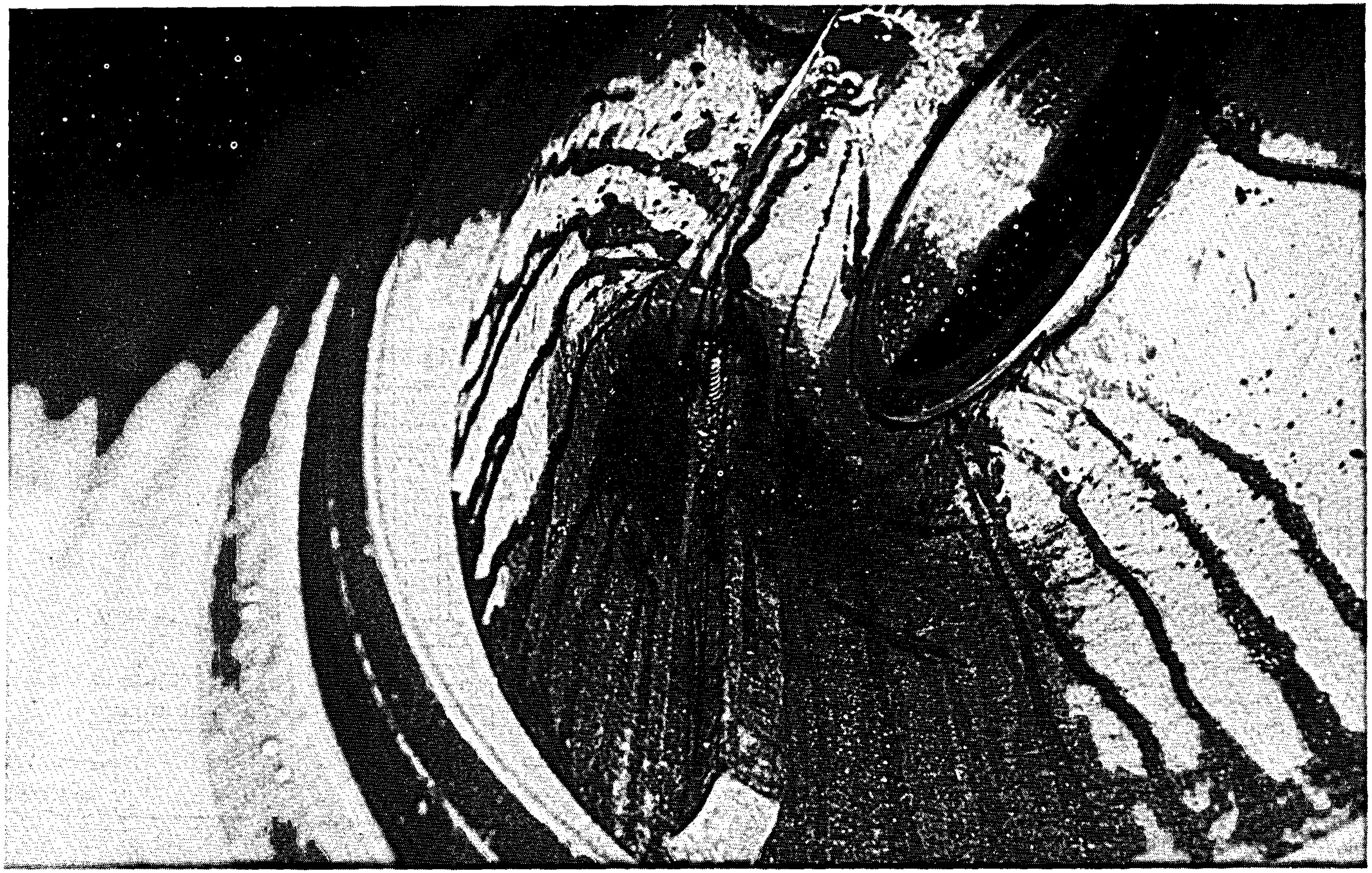

Figure 15. Typical valve damage observed.

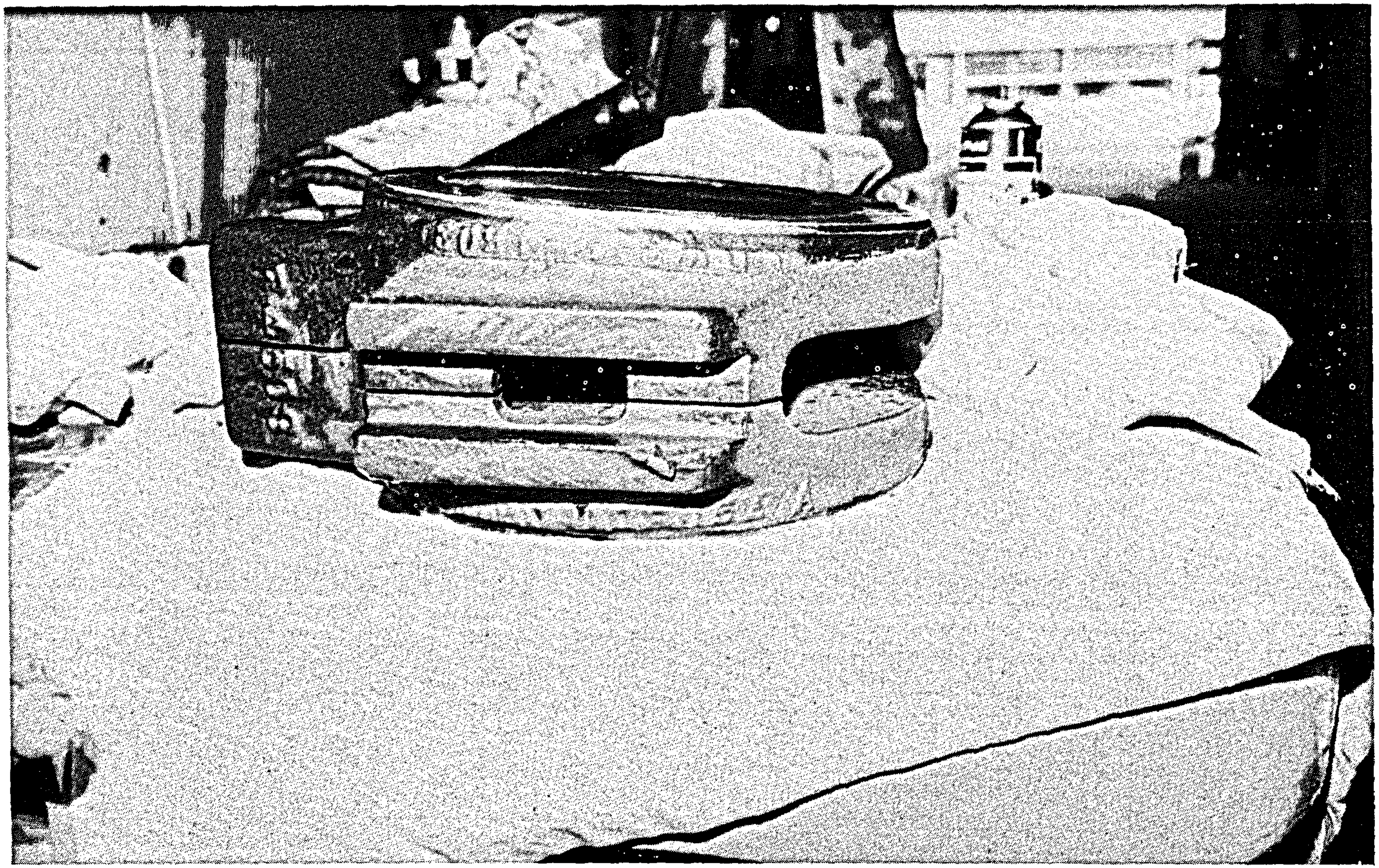

Figure 16. Typical valve damage observed.

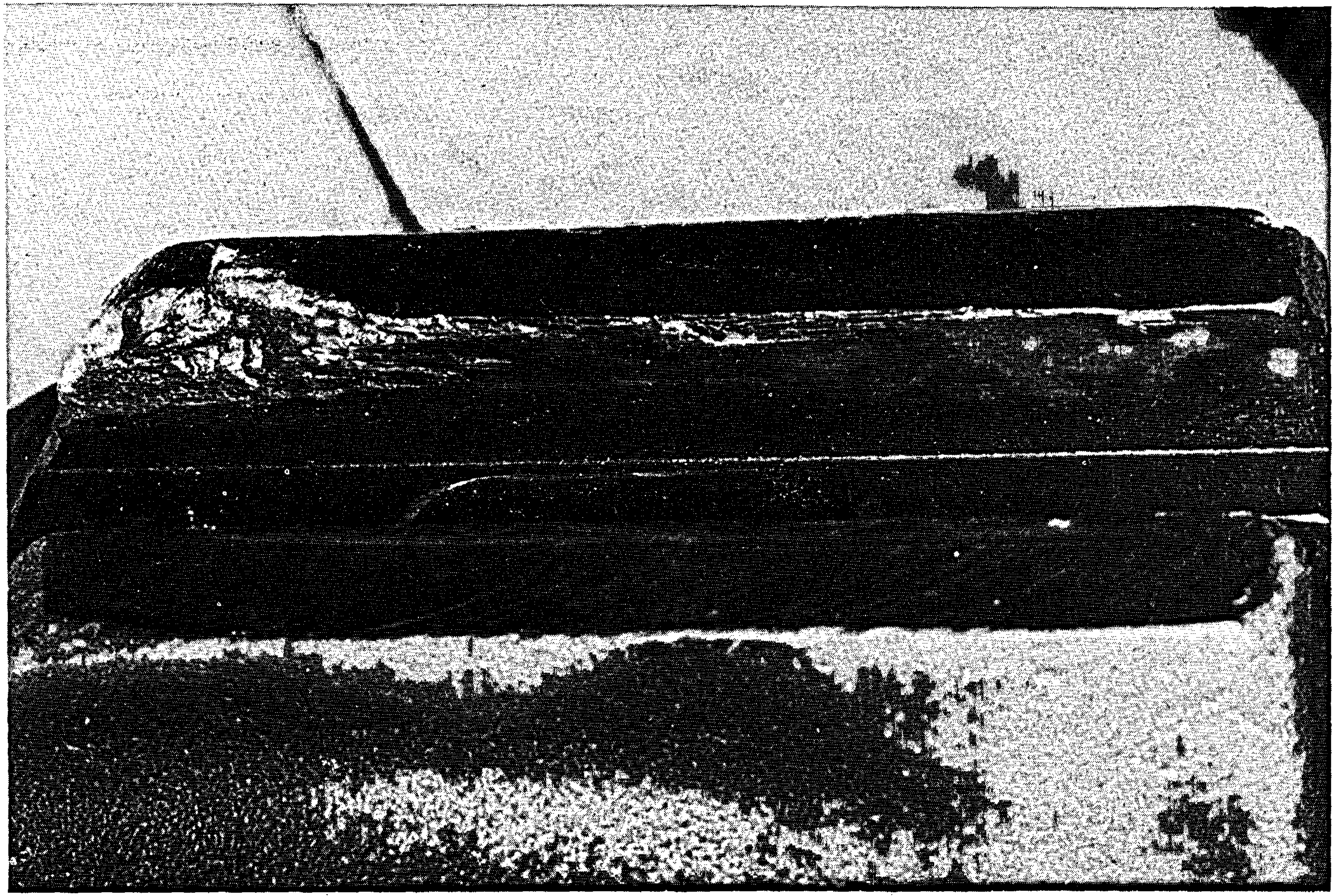

Figure 17. Typical valve damage observed.

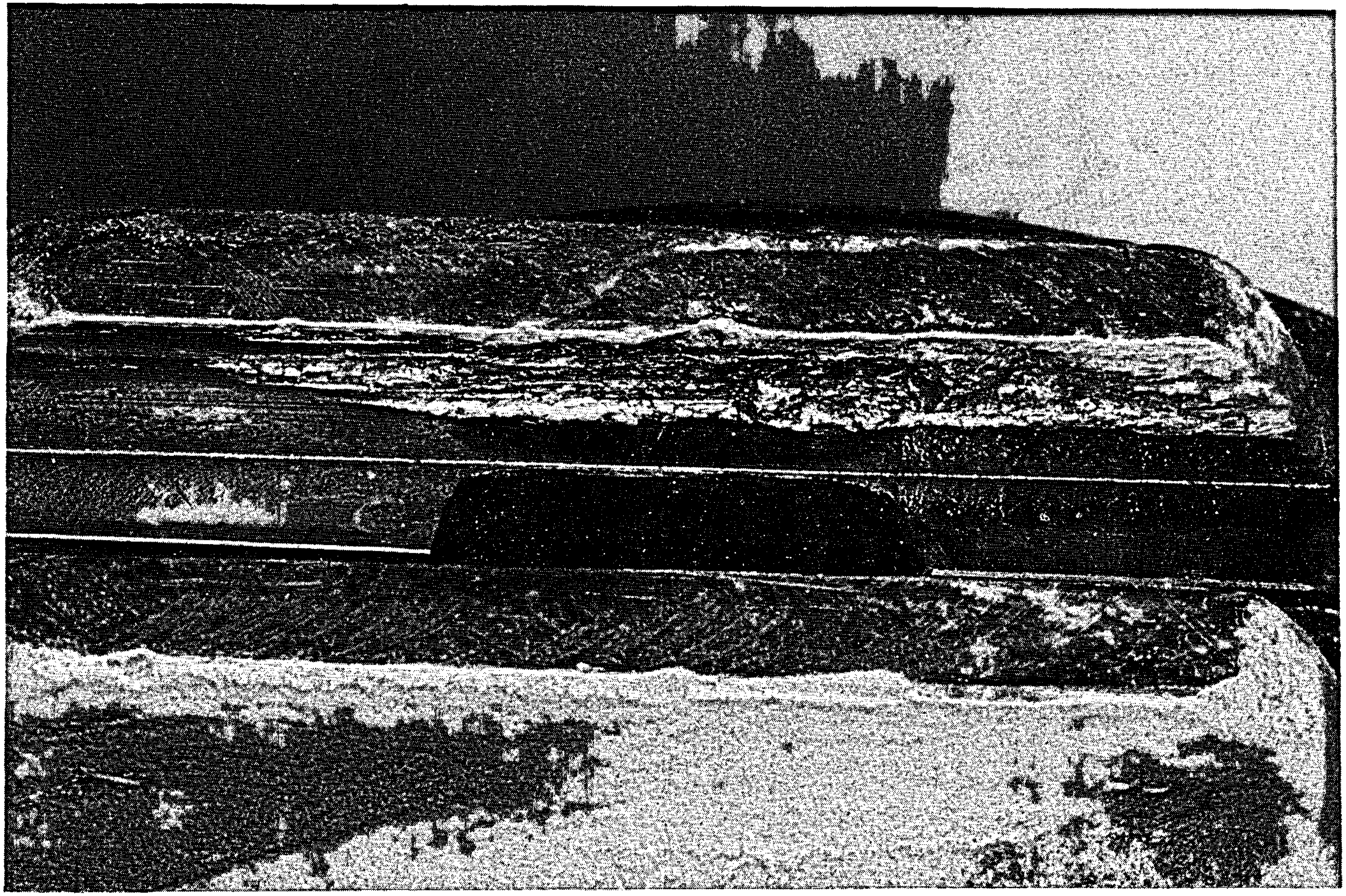

Figure 18. Typical valve damage observed.

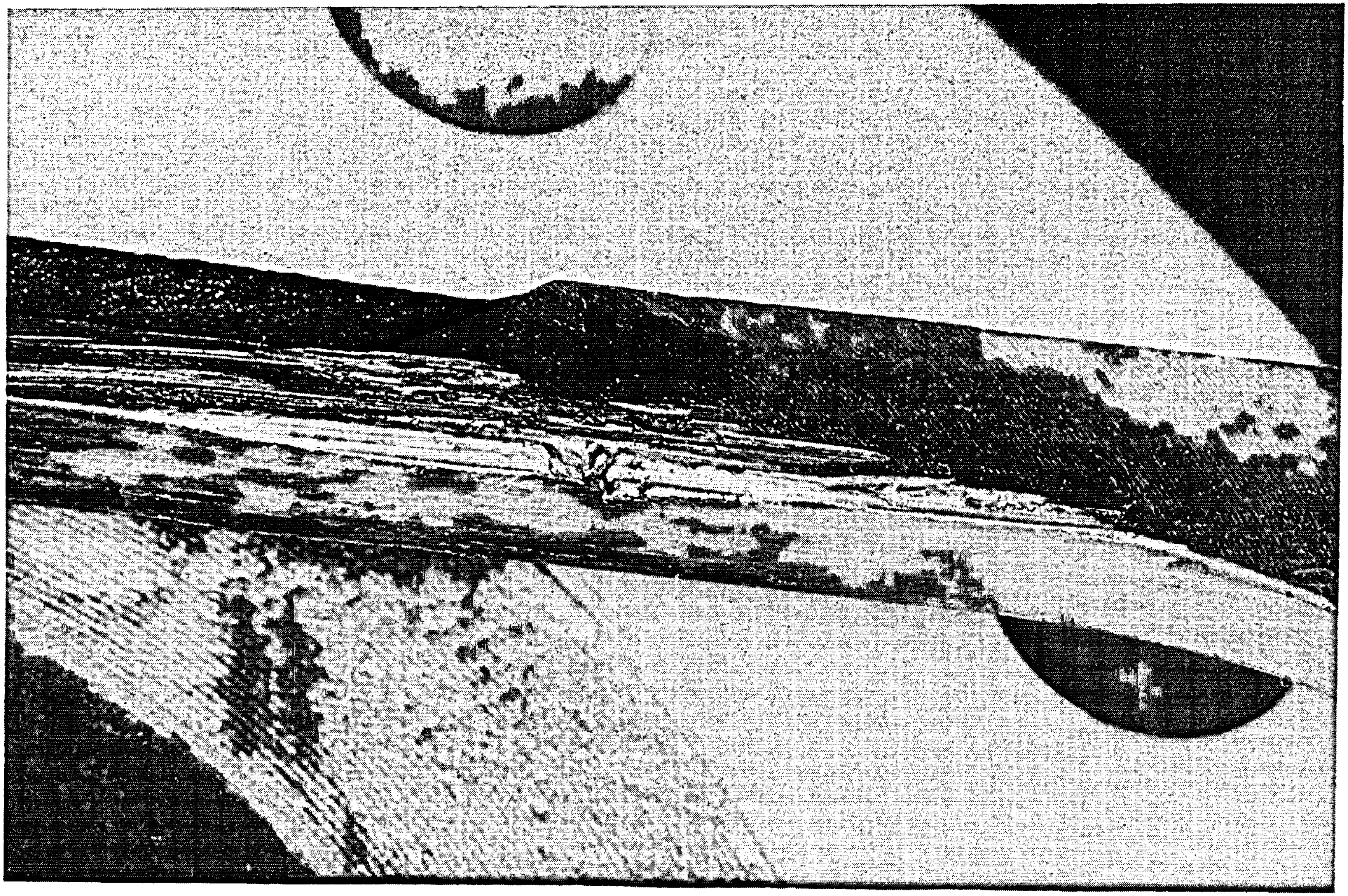

Figure 19. Typical valve damage observed.

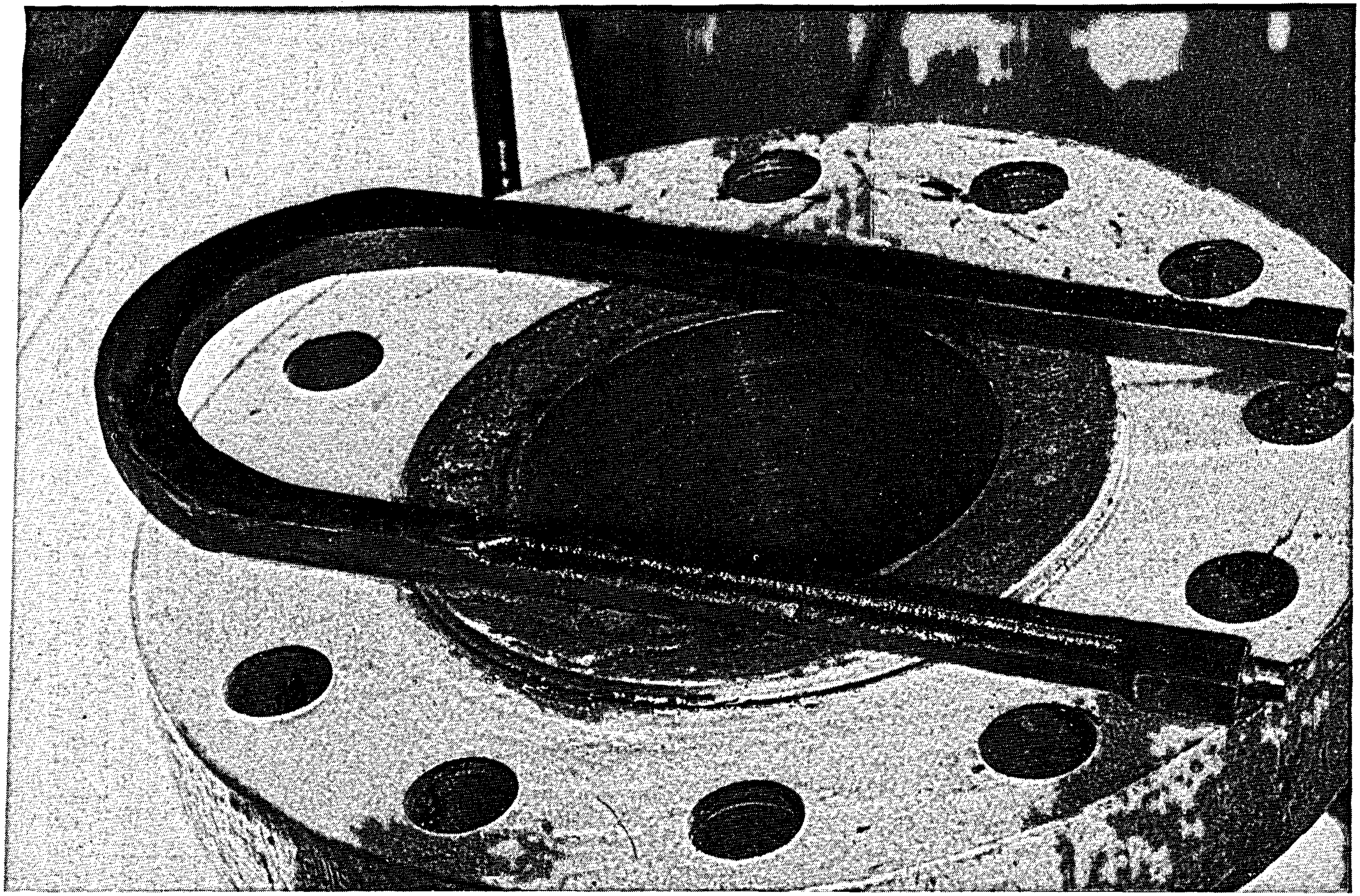

Figure 20. Valve 3 construction details.

At about the 10.5-s time line, the disc started to shave the seat. At about the 13-s timeline, the disc started to chatter across the seat, digging in and releasing twice before final seating. Posttest inspection also revealed the lower portion of the welded-in body guides had plastically deformed downstream. Valves 2 and 6 were manufactured by the same company and were basically the same design internally, but the stem force histories of the two valves were different. Valve 2 fell into the predictable category, while Valve 6 fell more into the unpredictable category.

Valves 1 and 4 were also from the same manufacturer. Both were in the unpredictable category, but the damage mechanisms were different. For Valve 1, the primary damage was to the seat. For Valve 4, the guide surfaces showed more damaged. The type of damage experienced by Valve 1 (Valve A in Phase I tests, where the flow through the valve was reversed from the Phase II direction), was dependent on flow direction. In Phase I, the damage was heavier on the guide surfaces.

The dissimilarity in stem force histories between valves of different sizes and valves of the same size with reverse flow, and the fact that valve stem force predictability appears to be detectable only by testing will make it difficult to accurately evaluate valve test data for other valves.

4.3 Parametric Studies

Valves 2 and 3 were both in the predictable category, and, as seen from the stem force traces, did not suffer extensive damage during the first flow interruption test at NOP/NOT. These valves were then subjected to flow interruption tests at higher and lower pressures and temperatures to determine the influence of pressure and fluid properties on valve stem force.

If all of the parametric studies could have resulted in just one parameter being varied, then the tests could have been compared to each other to determine the effect of the parameter (e.g., fluid properties). That was not the case, however. Calculating the exact temperatures and pressures in the test loop accumulators to provide the exact conditions at the valve could not be verified. This fact, along with other facility limitations (e.g., volume) resulted in tests that cannot be compared without some type of normalization. An example of this is shown in Figure 21, where Valve 2 was subjected to four different fluid states at the same initial target pressure of 1000 psig.

The individual stem force histories for the four tests plotted in Figure 21 are very different in shape and their comparisons to the calculated stem force histories are very different. The calculated stem force history with the 0.5 constant disc factor bounds the steam test stem force history and it marginally bounds the 10°F subcooled test; however, it does not bound the other two subcooled test stem force histories. Note also that as the degree of subcooling is increased for each of the tests, the valve stem force history at flow initiation becomes increasingly more positive. In fact, the cold water stem force history goes above the zero stem force line at the initiation of flow, representing tension instead of compression in the stem. The stem force histories also appear to be affected by fluid properties and possibly by forces other than those identified in Equation (1). The stem force histories in Figure 21 cannot be directly compared because the actual test pressures vary from the target pressure.

Our first attempt at normalizing the test results was to use Equation (1) to solve for the disc factor of each stem force history shown in Figure 21. This comparison is shown in Figure 22. The plot is read from right to left as the valve closes. As time increases, the disc factor increases in the negative convention and is plotted against stem position. The zero stem position represents flow isolation. At flow isolation, the area and the ΔP terms in Equation (1) become constant. From 0 to -10%, the stem travel involves seating and wedging, and there are no terms in the equation to represent these resistances. The calculation must be conservative enough to bound these additional loads. The figure indicates that the disc factor is influenced by fluid properties, steam being the best performer and cold water the worst. This is contrary to the expectation that water should be a better lubricant than steam.

Our next effort was to determine if the disc factor was pressure- dependent. Figure 23 shows this comparison for Valve 2 using the three parametric tests where the fluid properties remained constant and the pressure was varied. At flow isolation, the

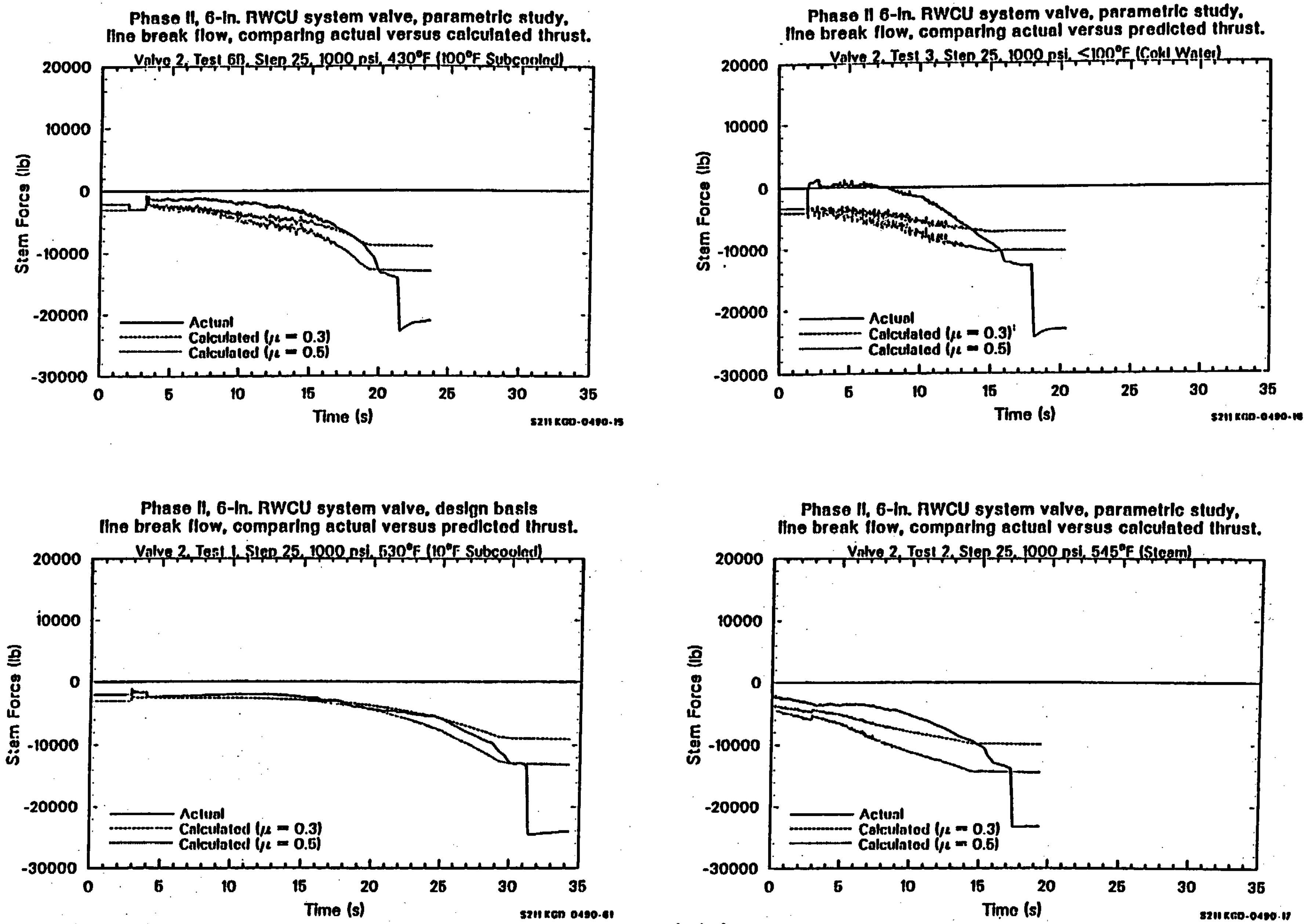

Figure 21. Valve 2 closing at full flow. Effect of subcooling on break flow isolation.

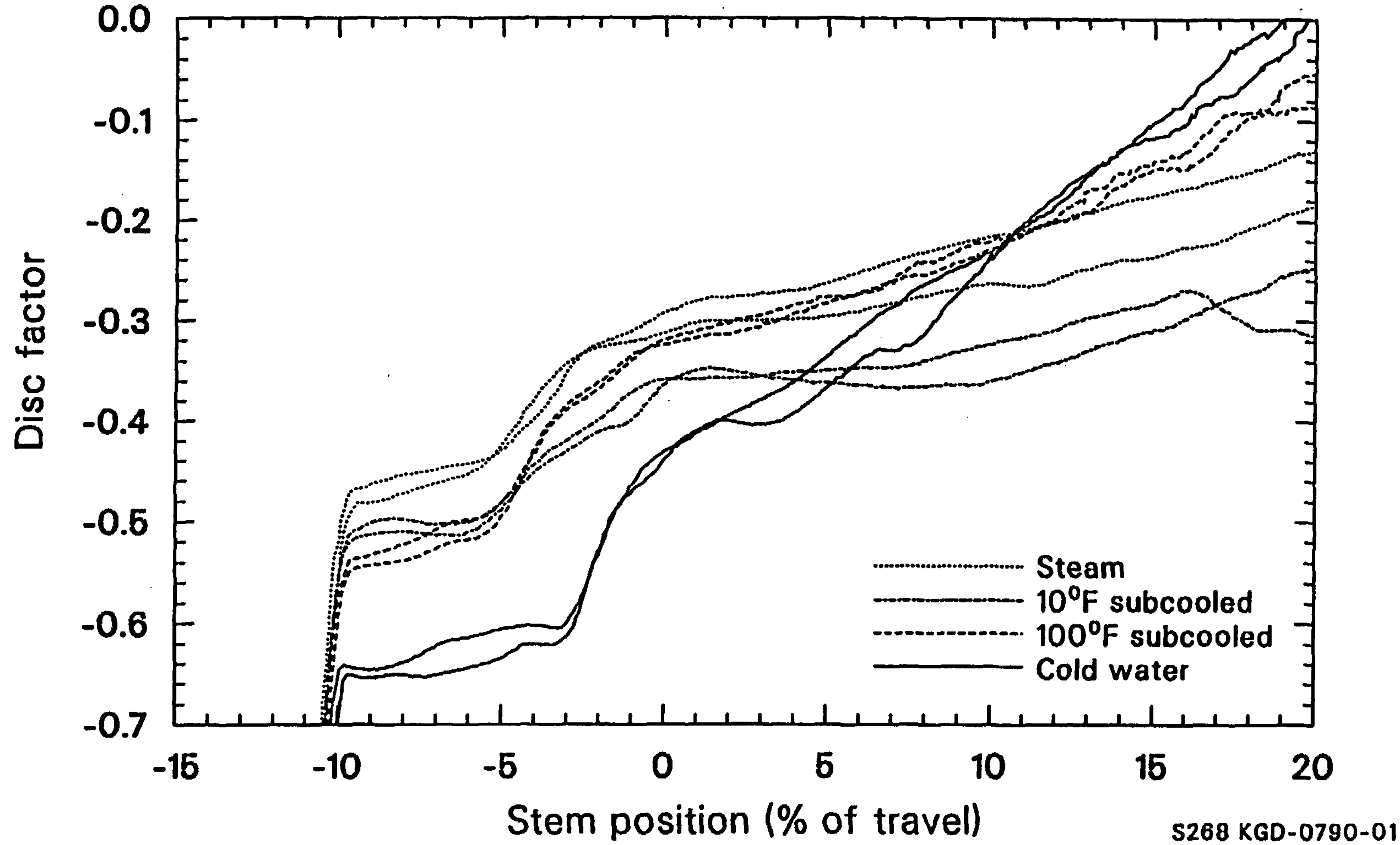

Figure 22. Valve 2 closing at full flow. Effect of subcooling at 1000 psig on the disc factor.

disc factor was not pressure-dependent, though there was a small spread in the opposite direction from what one might expect. Here the 1400-psig test had a lower disc factor than the two lower pressure tests.

Figure 24 again shows Valve 2 with the fluid properties remaining constant and the pressure being varied. In time, the plot reads from left to right as the valve opens. The spread is not large at isolation (zero stem position) but the 1400-psig test has the lowest disc factor. This is not what one would expect, nor would one expect the disc factor to increase as the valve is opened further. Comparing Figures 23 and 24, the magnitude of the disk factor at isolation in the closing direction is 0.3 to 0.35, whereas at opening the disc factor at isolation is 0.5 to 0.55. Unless there are mechanical effects involved, this is inconsistent with Equation (1). The opening stem force assisted by the stem rejection load should not cause the disc factor to increase.

Our next effort was more basic. We took the ratio of stem force divided by the differential pressure for Valve 2, where the fluid properties were constant and the pressure was varied in both the opening and closing directions. Figure 25 represents this ratio for closing (read right to left) and Figure 26 for opening (read left to right). At flow isolation, the ratios are nearly the same, both for opening and closing. Again, this should not be the case if Equation (1) is valid because the stem rejection load is always out of the valve. Valve 2 has a 1.75-in. diameter stem, which for the 1000-psig test represents a 2410-lb stem rejection load, 20% of the stem force required to isolate flow. If the stem rejection load calculation was sufficient, we would subtract two times the stem rejection load from the closing stem force to get the opening stem force. Figures 25 and 26 indicate that this is not the case.

We performed this same analysis using Valve 3. Figures 27 and 28 show the closing and opening stem forces divided by ΔP for the constant degree of subcooling tests performed with this valve. The conditions for these three tests were actual BWR RWCU conditions of 10°F subcooled with the pressure varied from 0.667 to 1.5 times the normal operating pressure. Note the same kind of pressure distribution seen with Valve 2 and the same reverse order on opening. The 600-psig test required more stem force per pound of ΔP than the 1400-psig test. The closing traces are slightly more muddled but show the same general tendencies. This valve had a 1.5-in. diameter stem, and for the 1000-psig test should have had a 1770-lb stem rejection load, or more than a 40% swing between opening and closing. As with Valve 2, the stem rejection load, which is a significant percentage of a 6-in. valve opening or closing stem force, did not show up in the actual stem forces required to operate the valve. We know that a stem rejection load exists because we see it in the stem force load when the valve is pressurized prior to initiating flow.

Next we looked at Valve 5, a 10-in. HPCI steam valve. During the first high flow interruption test, there was some disc tipping and seat machining as a result. On subsequent cycles, the stem force histories looked much like those for other valves that only sustained minimal damage. Inspection of the valve internals showed that the disc machined the body sealing surface from about halfway closed to the mostly closed position, at which time there was enough disc-to-seat contact to straighten the disc. It appeared that after the initial machining of the conical surface, the wider seatface guided the disc similar to the performance of Valve 3, where its flexible guide was plastically deformed and the disc was guided by the downstream seat. This analysis is subject to other interpretations, but all of the evidence indicated that the valve, after the initial closing, performed as well as any of the other valves that were subjected to parametric studies.

The other two 10-in. valves sustained too much mechanical damage to be used for the thermal hydraulic studies. These two opening stem force histories do, however, support our analysis conclusions for Valve 5. Figures 29 and 30 show the closing and opening stem force divided by ΔP ratios for Valve 5. The first closing cycle at 900 psi does show the effects of seat machining. The others fall into a more normal pattern. The 10-in. valve at flow isolation was very much like the response of the 6-in. valves. The stem force divided by the ΔP ratio was roughly 25 compared to 9 for the steam test on Valve 2. This represents a response ratio of 2.63 for the 6- to 10-in. valves, while the port area ratio is 2.93. The 10-in. valve opening stem force ratio is quite different in comparison to the 6-in. valve opening stem force ratios. The highest stem force per pound of ΔP is not at seat lift-off, but comes later in the opening stroke. At 10% open, the stem force to open

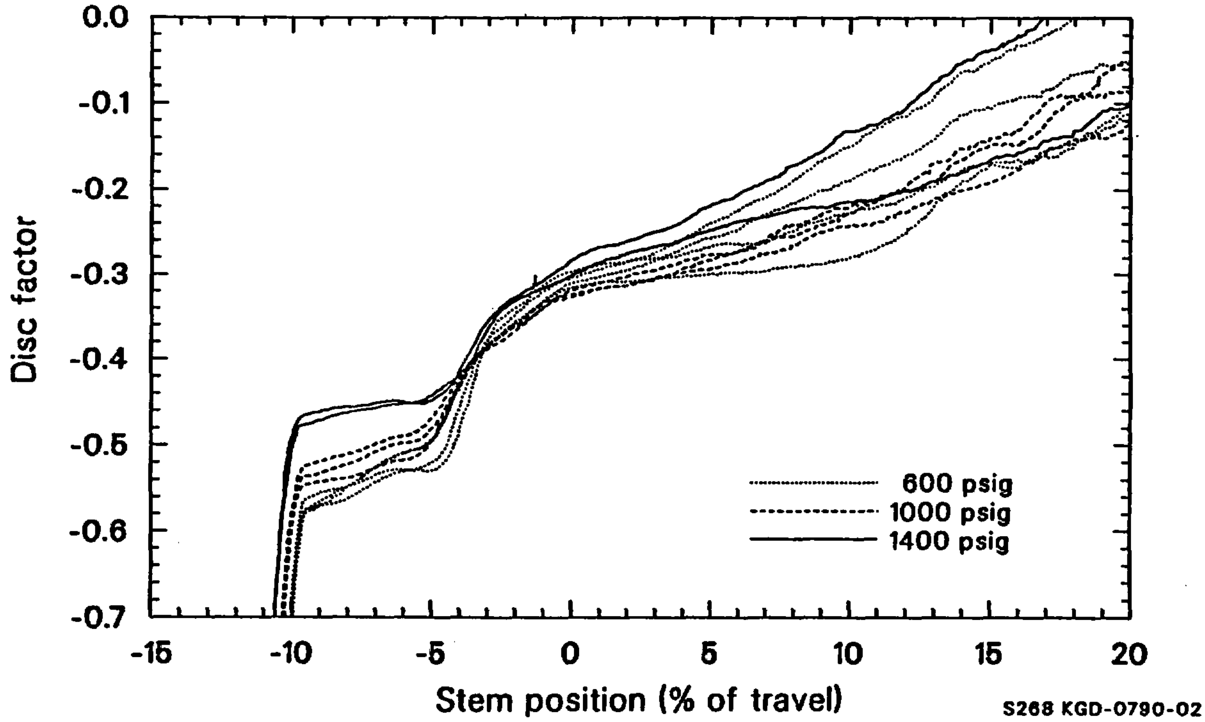

Figure 23. Valve 2 closing at full flow. Effect of pressure at 100°F subcooling on the disc factor.

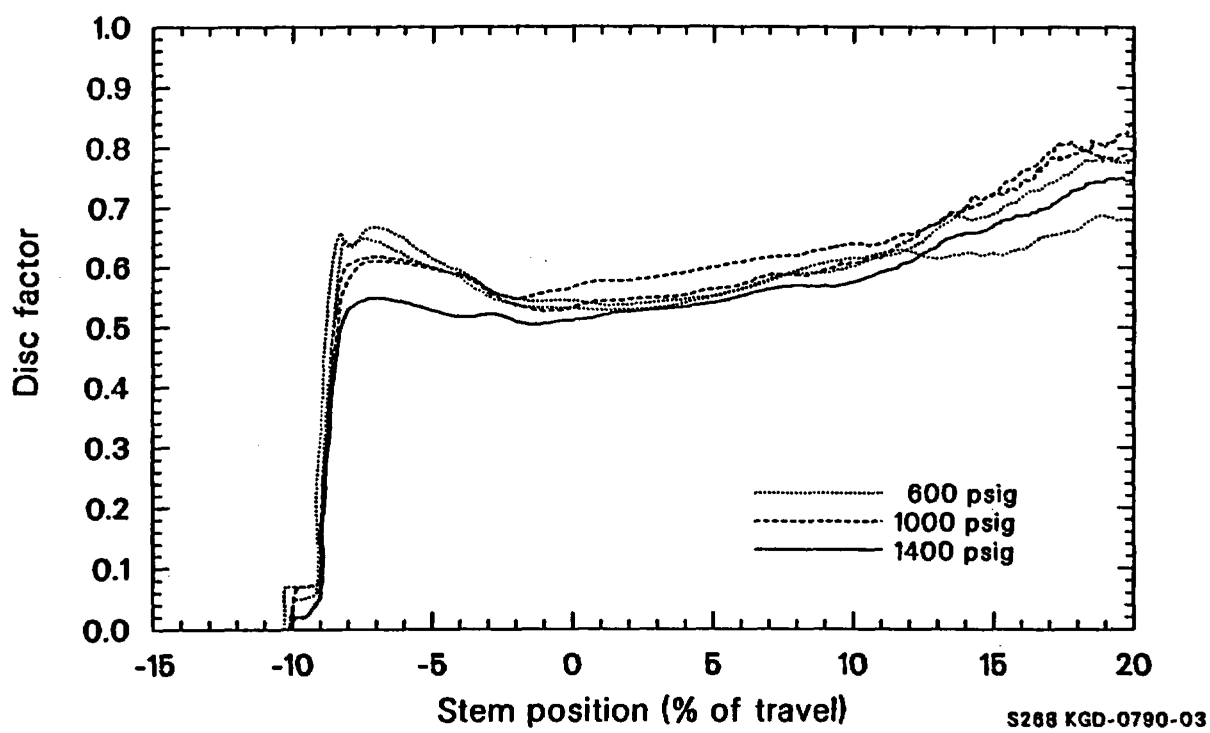

Figure 24. Valve 2 opening at full flow. Effect of pressure at 100°F subcooling on the disc factor.

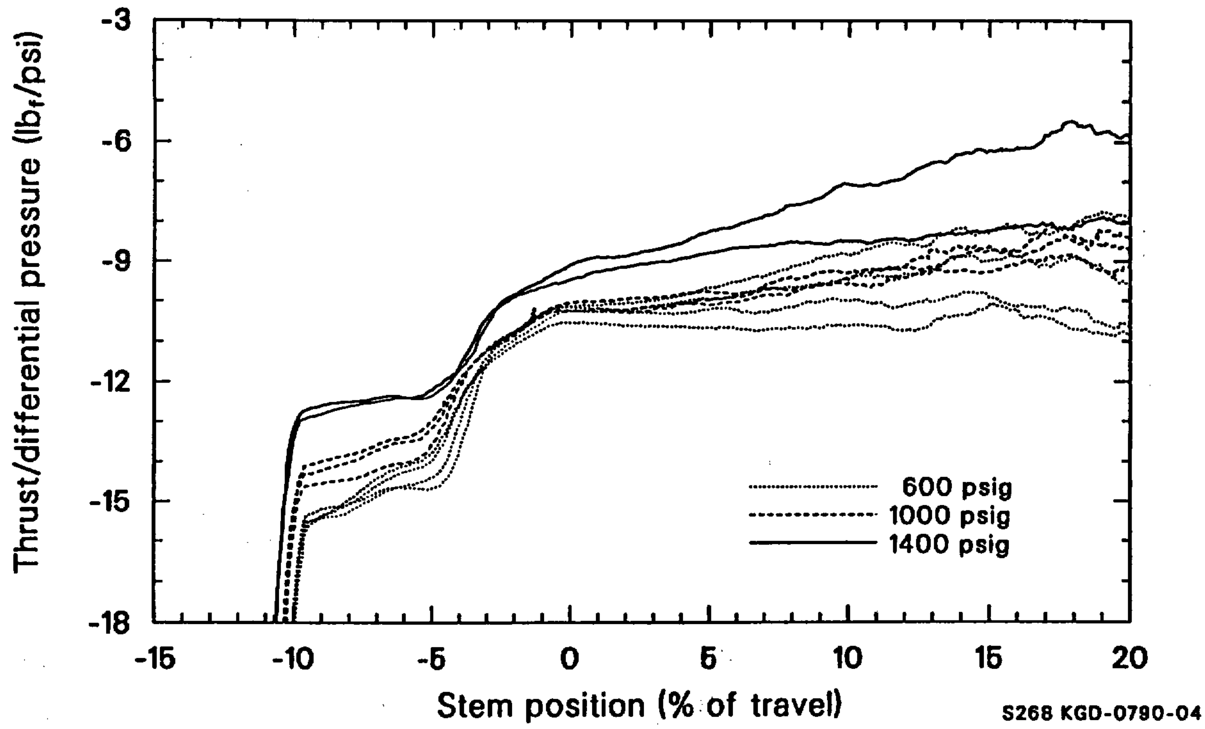

Figure 25. Valve 2 closing at full flow. Effect of pressure at 100°F subcooling on stem force/ΔP.

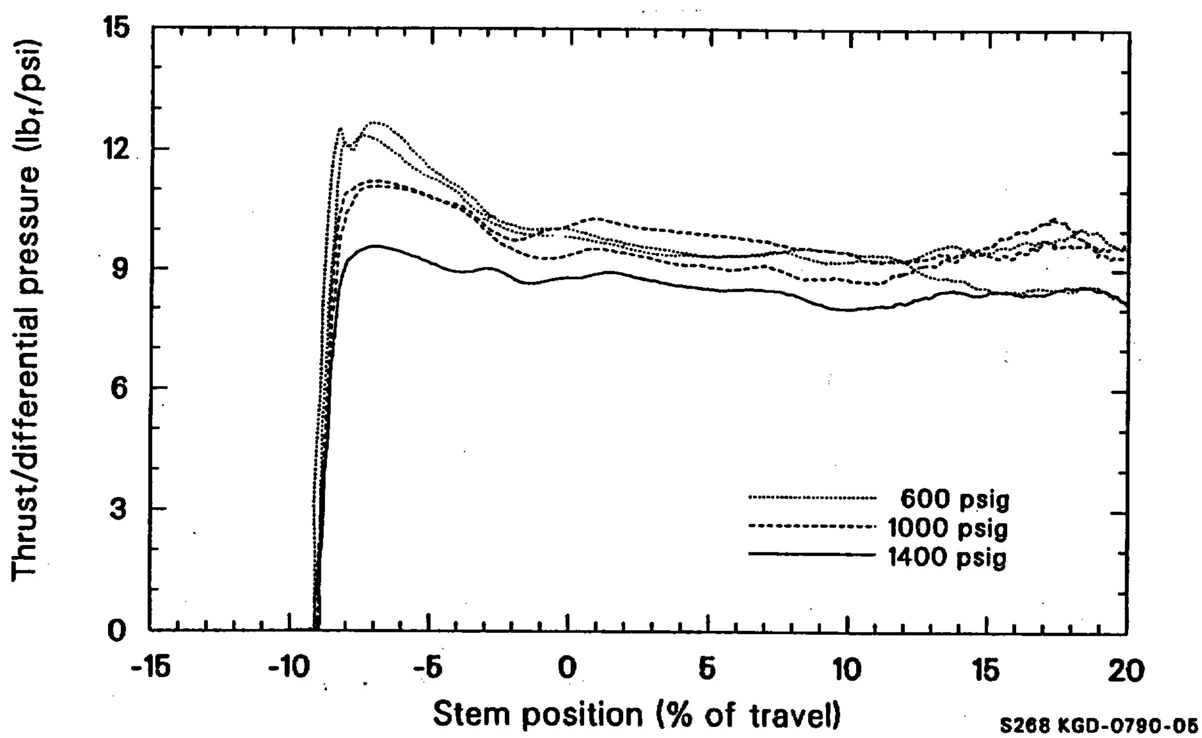

Figure 26. Valve 2 opening at full flow. Effect of pressure at 100°F subcooling on stem force/ΔP.

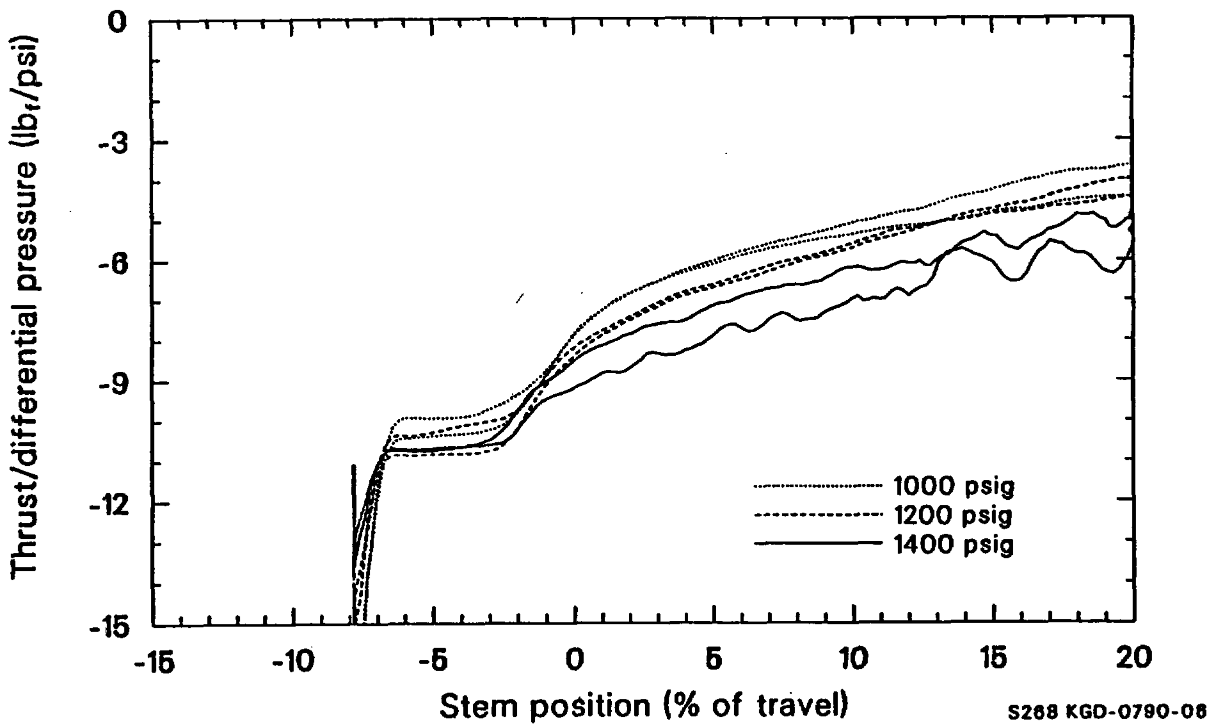

Figure 27. Valve 3 closing at full flow. Effect of pressure at 10°F subcooling on stem force/ΔP.

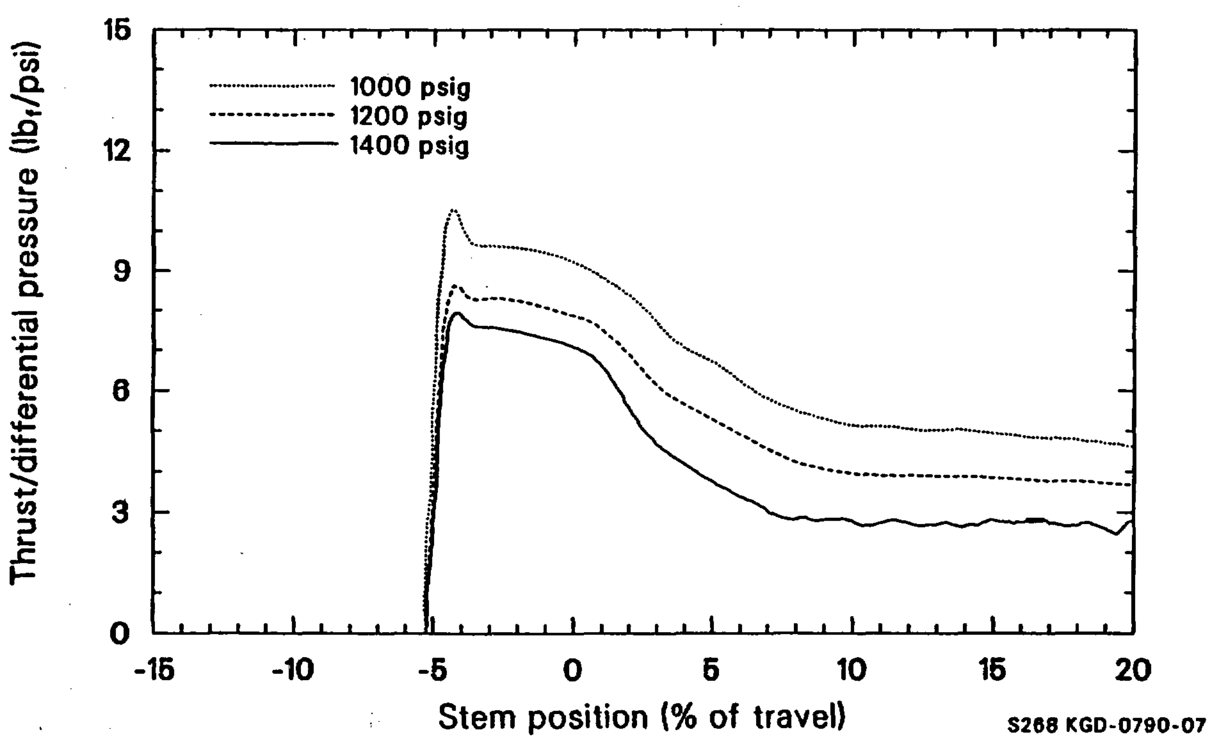

Figure 28. Valve 3 opening as full flow. Effect of pressure at 10°F subcooling on stem force/ΔP.

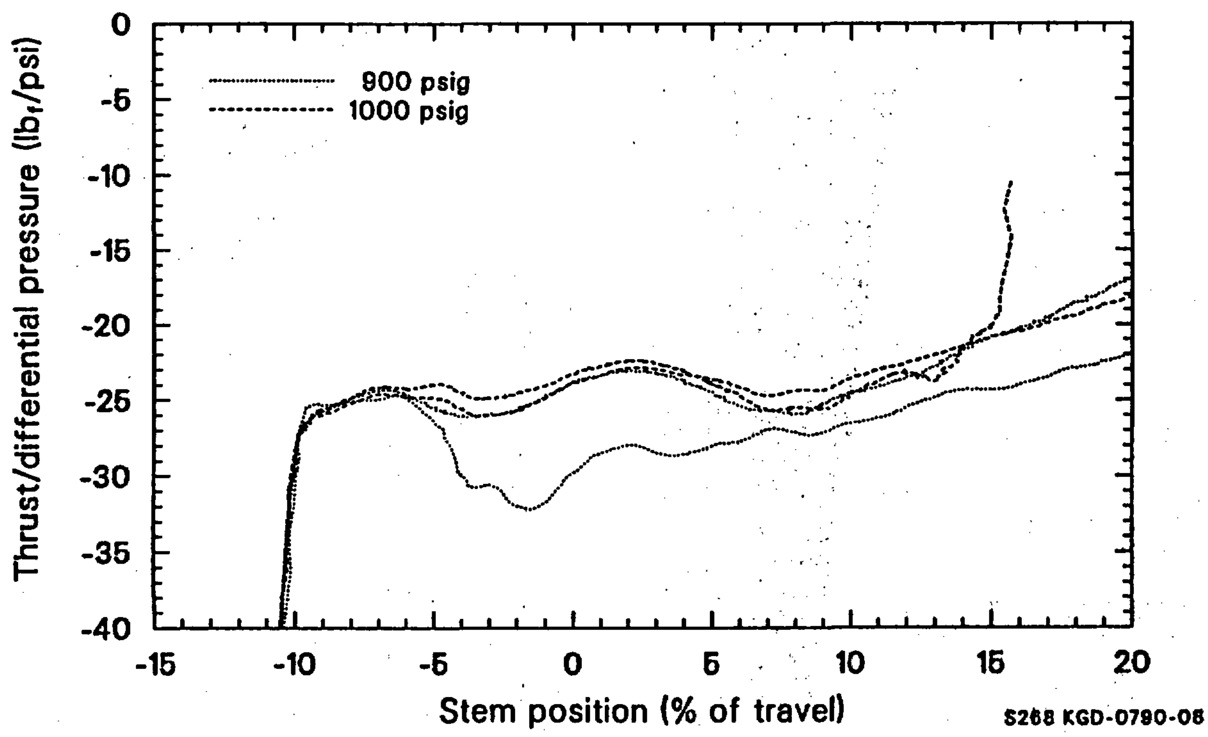

Figure 29. Valve 5 closing at full flow. Effect of pressure at saturation on stem force/ΔP.

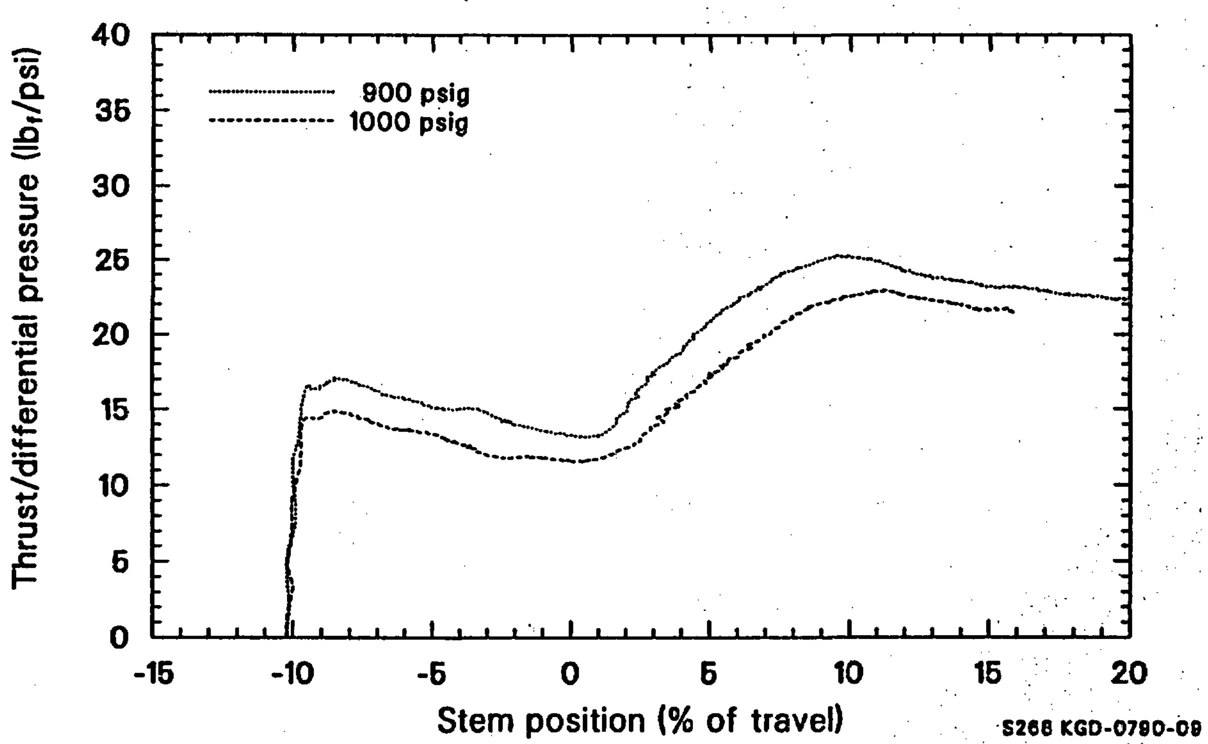

Figure 30. Valve 5 opening at full flow. Effect of pressure at saturation on stem force/ΔP.

equals the stem force to close at isolation. The other 10-in. valves show this same phenomenon, but the 10-in. valve response still does not show the stem rejection load effects that we would expect from Equation (1).

Next, we will look at our final parametric study where the pressure, within the capability of the facility, was constant and the fluid properties were varied. This study again was performed on Valve 2, one of the 6-in. valves. Figures 31 and 32 show the closing and opening stem force divided by ΔP. Within a given fluid, the opening and closing ratios are close at flow isolation. However, the responses to the fluids are very different. Steam has the lowest stem force-per-psi ratio, cold water the highest. The cold water opening test also shows the same stem force response characteristics as the 10-in. valves, where the highest stem force is not at unseating but later in the opening stroke.

The cold water closing stroke also has a unique shape. Figure 33, the actual stem force history for this closing, shows this response. The system is pressurized at the zero line, and from zero to 2-s the effects of a 1500-lb stem rejection load are seen. At flow initiation, the 2-s time line, the stem force takes a positive response. Because of facility capacity limitations, the valve was 40% closed prior to flow being initiated, so the disc was well into the flow stream. The valve was commanded to close at about the 3-s time line. We saw a small reduction in the tension load here, but it remained over the zero stem force line for the next 5 s of valve travel. The magnitude of the stem force history deviation from the calculated stem force histories led us to our first real breakthrough in understanding what was occurring inside of the valve. Figure 34 shows the details of the areas where pressure forces can act on the disc and stem and where we drilled the three pressure measurement ports into the valve bodies for pressure distribution studies.

Review of the bonnet, upstream, under disc, and downstream pressures, and the various internal valve surfaces where these pressures can act has started to provide some understanding of the valve behavior outside of the loads accounted for in Equation (1). Prior to flow initiation, a stem rejection force as predicted by Equation (1) was observed. However, after the initiation of flow, the stem rejection component always decreased and, in the case of cold water, reversed its direction. Closer investigation of this phenomenon revealed that before flow initiation, the pressures throughout the valve were identical. Thus, the pressures acting on top of the disc that exerted a downward stem force were identical to those pressures acting on the bottom of the disc that exerted an upward stem force. The upward and downward stem forces did not balance, however, because the areas on the top and the bottom of the disc were not the same, differing by the area of the stem. Thus, the classic stem rejection term in Equation (1) is correct when the internal pressures are identical.

When flow started, however, pressure imbalances developed within the valve, which affected the net force balance on the disc. The figures on the following pages present pressure ratios and provide insight into the pressure loading forces and how they affect valve performance. The first two figures show the Valve 2 ratio of the bonnet pressure to the under disc pressure versus valve stem position. Figure 35 presents the effect of pressure at a constant 100°F subcooling, whereas Figure 36 presents the effect of subcooling at a constant upstream pressure of 1000 psig. Notice that changing the pressure at a constant degree of subcooling had little effect on this pressure ratio, which remained relatively constant in the 1.03 to 1.08 range. However, varying the degree of subcooling had a very profound impact on the pressure ratio; steam and the 10°F subcooling liquid having the lowest ratio (1.02 to 1.04) and cold water having the highest (1.10 to 1.25).

The area on top of the disc was less than the area on the bottom of the disc, differing by the area of the stem. Thus, a bonnet-to-under disc pressure ratio in excess of 1.0 did not imply that a downward or self closing force existed. Instead, the pressure imbalance had to be large enough to overcome the area differential, which is the classic stem rejection term of Equation (1).

For Valve 2, the ratio of the area on the bottom of the disc to the area on the top of the disc was roughly 1.10. Although there were other pressures acting on the disc that affected the net force balance, a pressure ratio in the vicinity of 1.10 would have been necessary before a self closing force would have been expected. A smaller pressure ratio would have promoted a net outward force, but this force would have been much less than the classic stem rejection force.

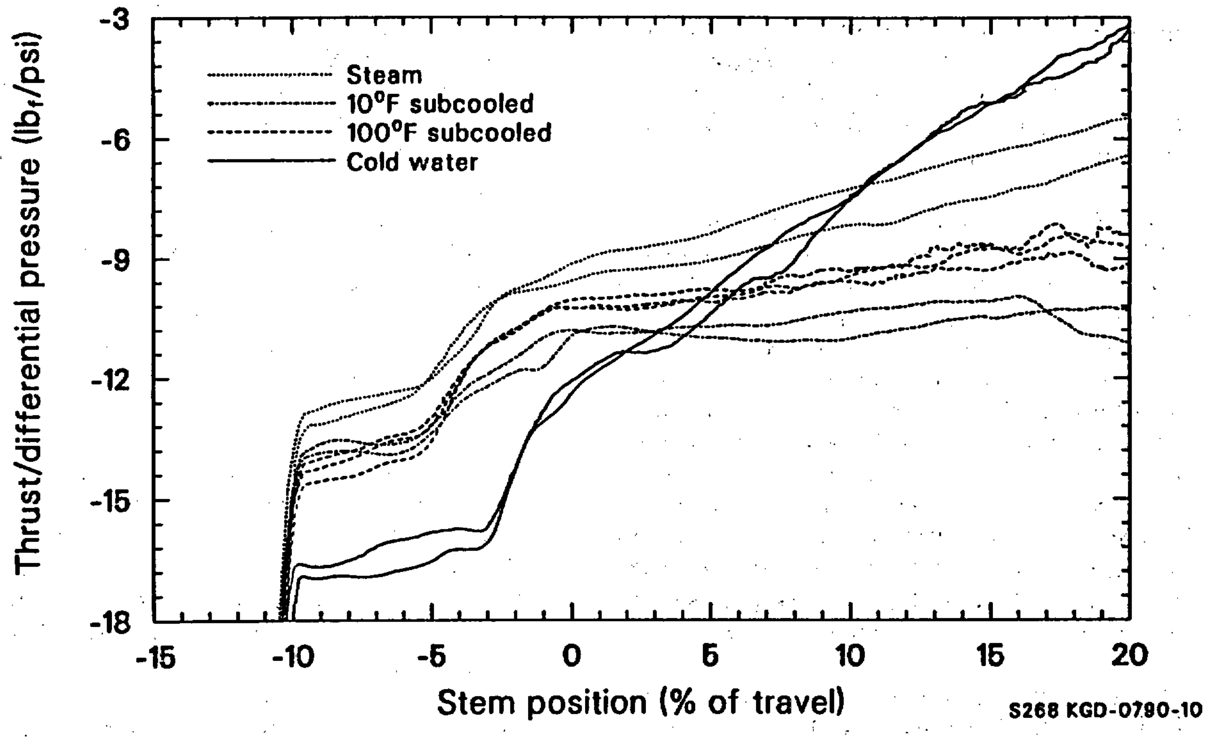

Figure 31. Valve 2 closing at full flow. Effect of subcooling at 1000 psig on stem force/ΔP.

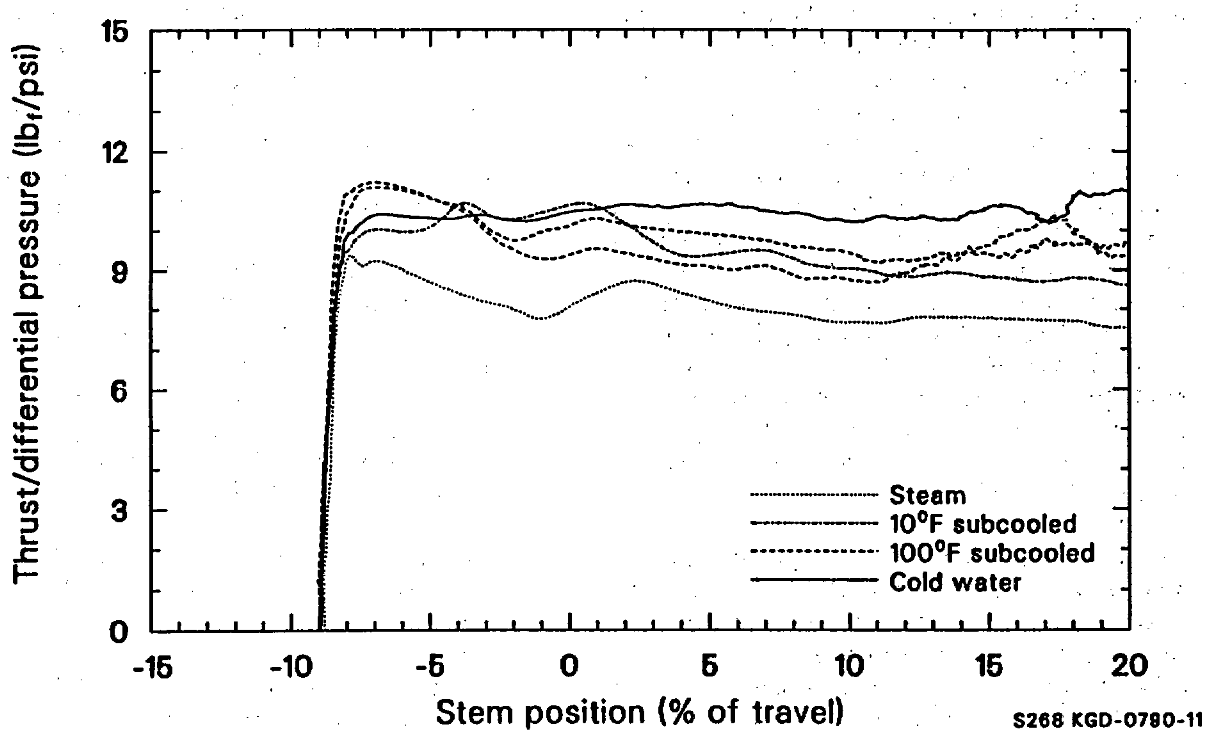

Figure 32. Valve 2 opening at full flow. Effect of subcooling at 1000 psig on stem force/ΔP.

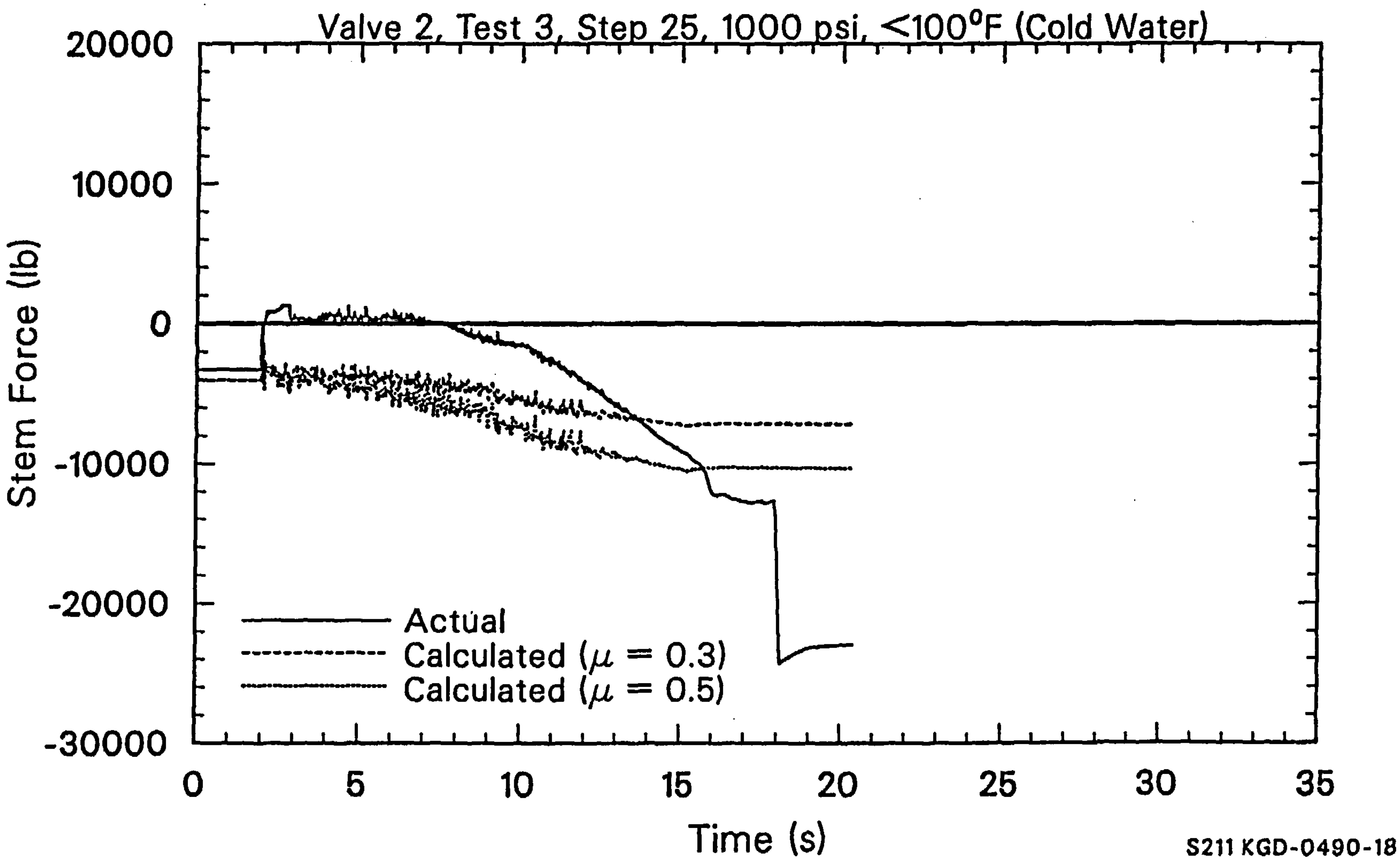

Figure 33. Valve 2 closing at full flow. Stem force at 1000 psig, cold water.

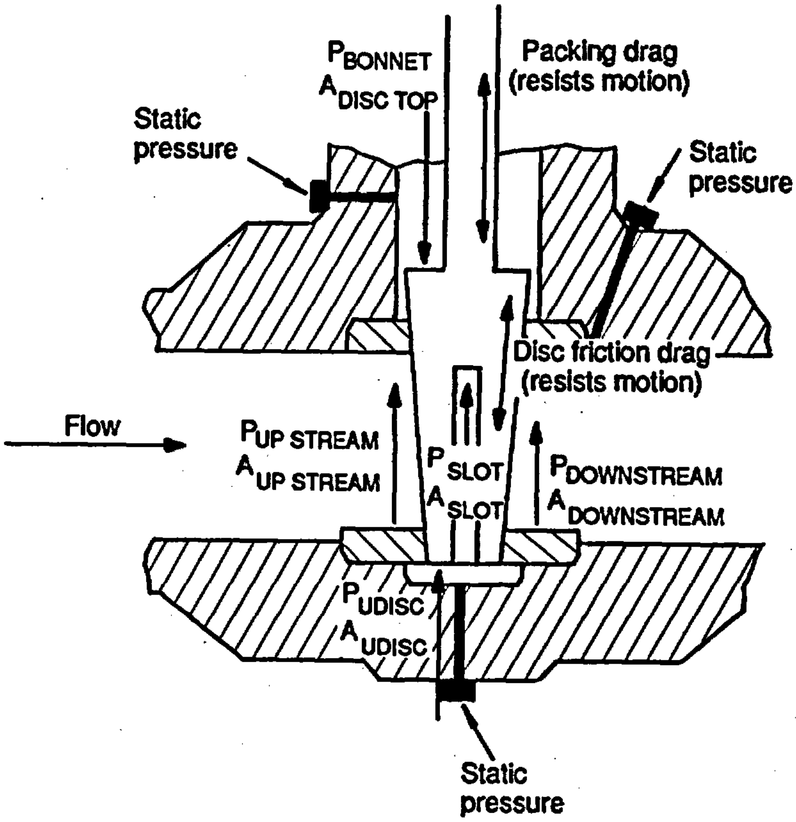

Figure 34. Valve disc cross section showing pressure loads.

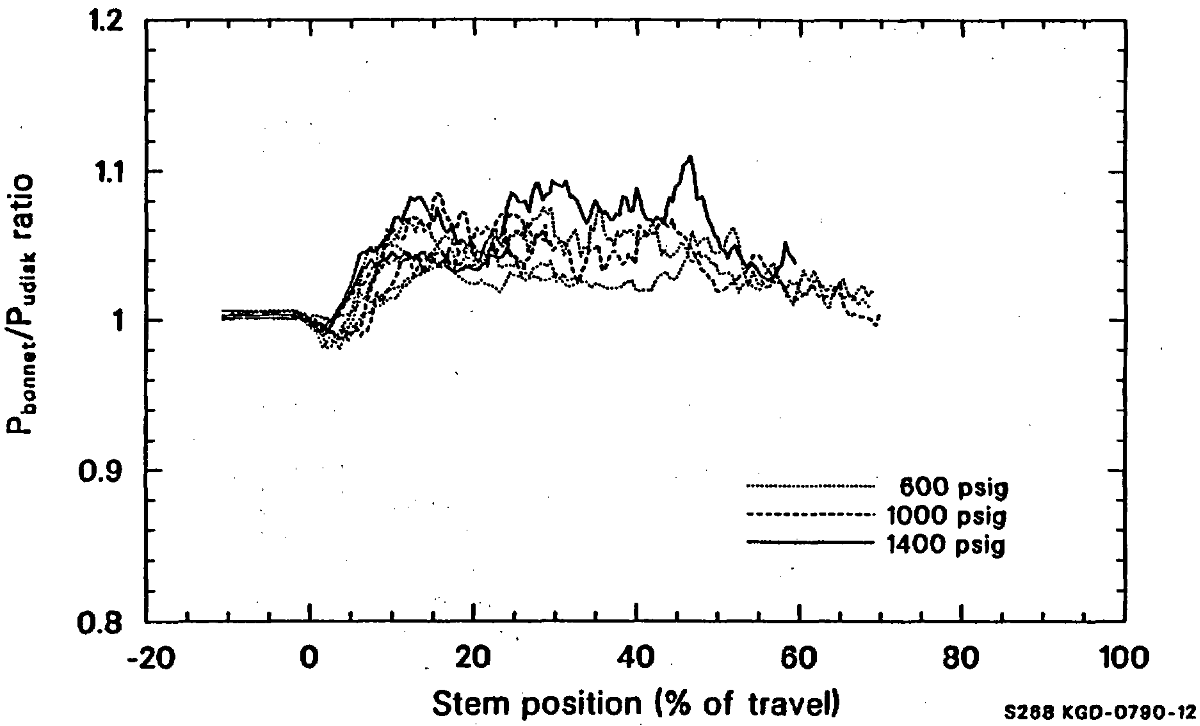

Figure 35. Valve 2 closing at full flow. Effect of pressure at 100°F subcooling on the bonnet-to-under disc pressure ratio.

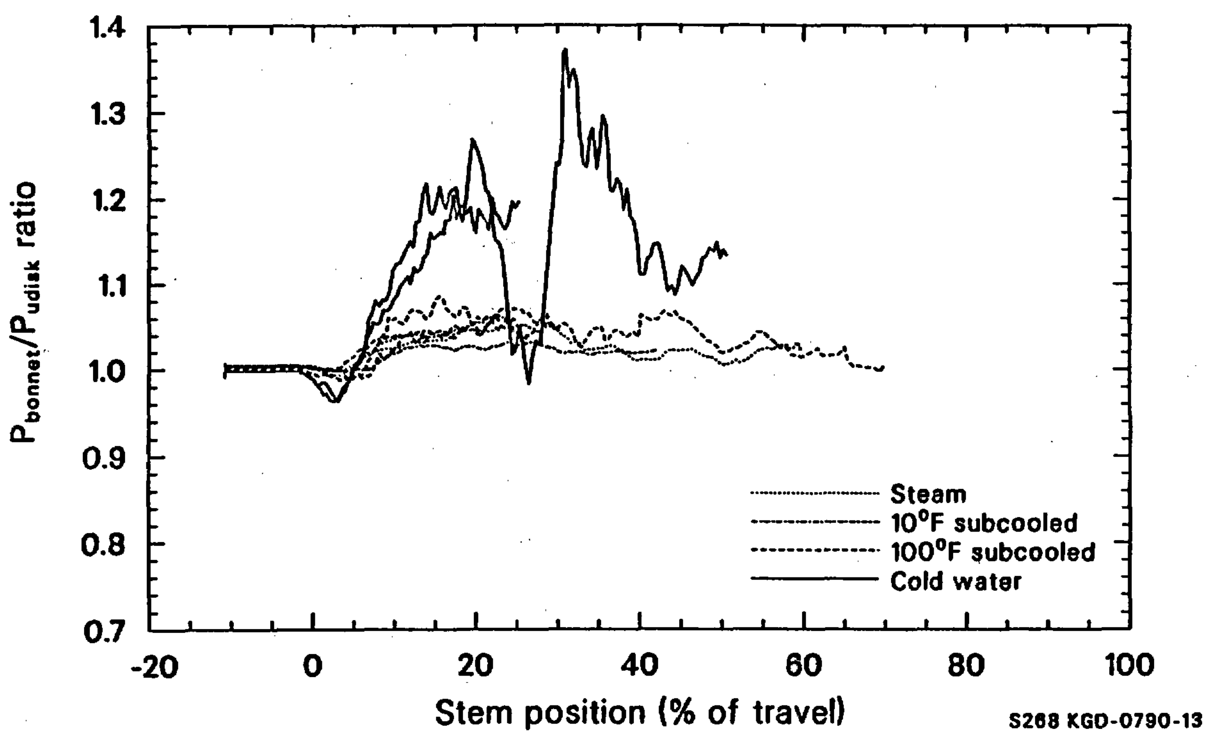

Figure 36. Valve 2, closing at full flow. Effect of subcooling at 1000 psig on the bonnet-to-under disc pressure ratio.

The effect of subcooling was very evident in other pressure comparisons throughout the valves. For instance, the pressure under the disc relative to the upstream pressure, as shown in Figure 37, displays the effect of fluid subcooling. The difference was the result of fluid acceleration through the valve and the expansion potential of the fluid state. For instance, as cold water accelerated through the valve, the pressure dropped. The fluid state had virtually no expansion potential and was also sufficiently removed from saturation so that flashing did not occur. Thus, this fluid state had the largest pressure drop in the throat region of the valve relative to the upstream pressure. Steam, on the other hand, could expand but had no flashing potential. Thus, the pressure in the throat region of the valve dropped relative to the upstream pressure, but to a lesser extent than cold water.

The 10 and 100•F subcooled states had the potential to both flash and expand. Thus, the pressure ratio did not drop to the extent observed with the cold water and steam fluid states. Furthermore, the 10•F subcooled fluid was closer to saturation than the 100•F subcooled fluid and thus flashed easier. As a result, the drop in the pressure ratio of the 10•F subcooled fluid was less than the drop of the 100•F subcooled fluid.

Conversely, Figure 38 reveals that the effect of pressure on the upstream-to-under disc pressure ratio was relatively nonexistent. Although there appeared to be a trend, closer examination revealed that all the pressure ratios were relatively constant in the 0.80 to 0.84 range. Any observed spread was more likely the result of slight variations in the actual subcooling of the fluid ratio than the result of a pressure effect. The other pressures monitored on other valves during the tests displayed similar trends relative to subcooling and pressure. As such, they are not presented.

Having observed several pressure trends in Valve 2, it was interesting to see if these same trends existed in the other valve pressure ratios. Remember, however, that parametric testing with the other valves involved pressure variations only. Only Valve 2 included a parametric study that addressed subcooling.

Figure 39 presents the bonnet pressure-to-under disc pressure ratio versus stem position for Valve 3. The ratios are indeed very similar to those observed for Valve 2 and again reveal no effect of pressure. The other pressure ratios also reveal trends similar to those observed in Valve 2. As such, they will not be presented.

Figure 40 presents the bonnet pressure-to-under disc pressure ratio for Valve 5. Remember that this was a 10-in. valve, whereas the previous valves were 6-in. Although this pressure ratio exceeded 1.0, the magnitude was much less than that observed with the smaller valves. The pressure ratio for 10-in. Valve 6 (Figure 41) displayed similar results. These larger valves had larger internal clearances than the smaller valves. Such differences may have affected the internal flow path resistances, flows, and the resultant flow and pressure distributions. As such, the resultant performance of a valve may not have been truly scaled to size. The pressure ratios discussed earlier are virtually identical to the trends presented with Valve 2. As such, they will not be presented.

We have not made all the physical valve measurements necessary to create a complete investigation. However, it is apparent from the responses of the valves during the various parametric studies that Equation (1) is incomplete and that the missing terms have a first-order effect on the observed responses of the valves.

Although not quantitatively defined to date, the loads that will most likely be included in a more complete stem force equation are disc drag, packing drag, a net force balance on the disc (top to bottom), and fluid properties effects.

4.4 Other Concerns

Test results from the gate valve test program have shown there were two classes of valves and valve responses to the flow interruption test loads: valves bounded by the disc load calculations (albeit at a higher disc factor than previously used by industry) and those damaged by the loading. The amount of damage and the resulting loads were valve-specific, depending on the stack up of internal valve tolerances and clearances, which we consider nonpredictable even with a large disc factor to cover the unknowns.

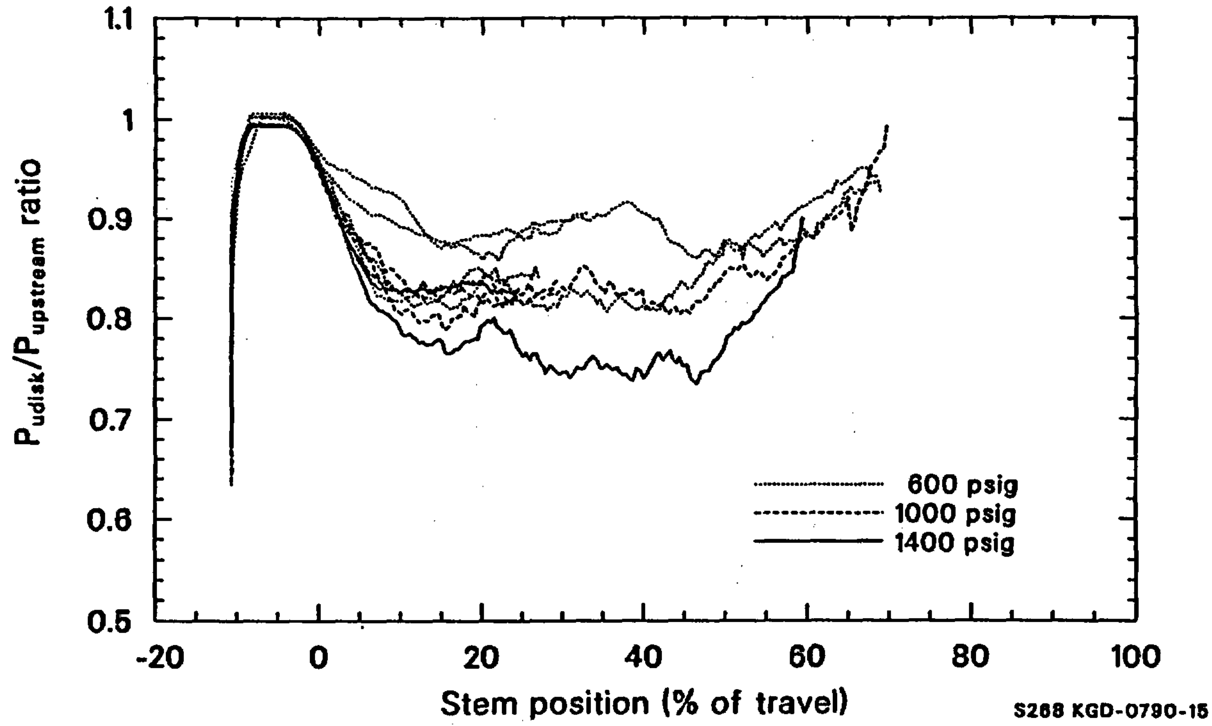

Figure 37. Valve 2 closing at full flow. Effect of pressure at 100°F subcooling on the under disc-to-upstream pressure ratio.

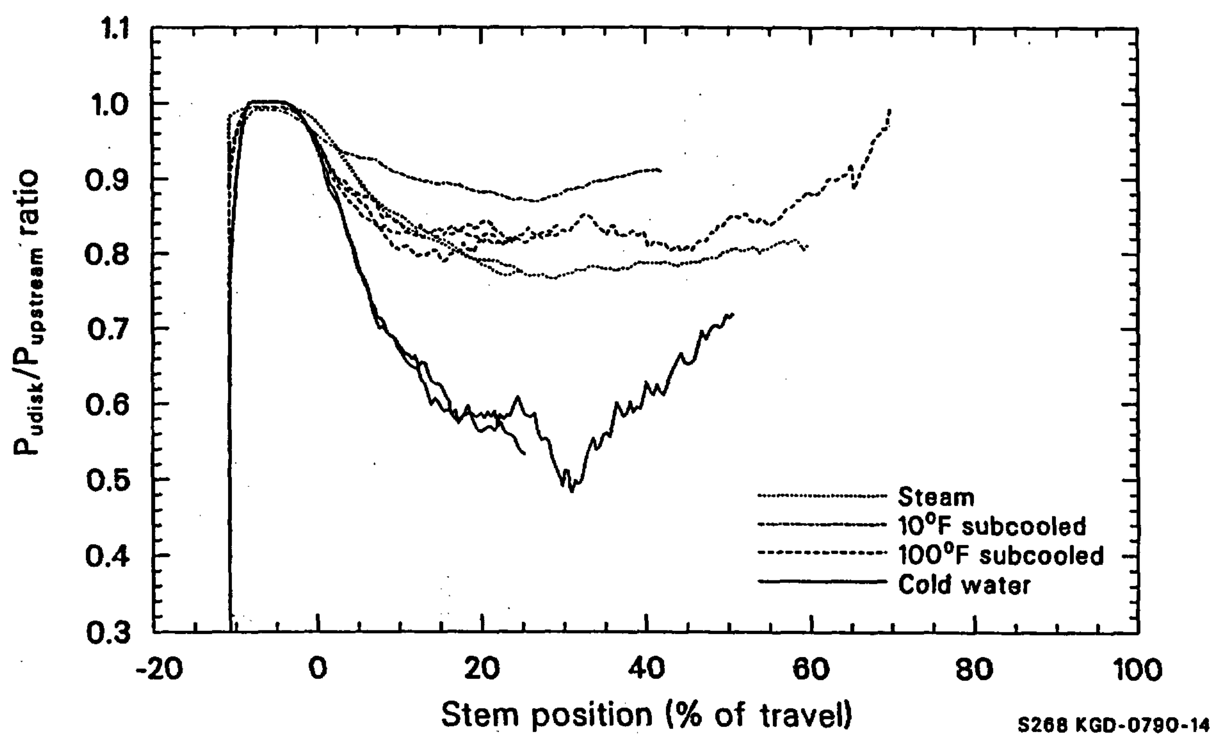

Figure 38. Valve 2 closing at full flow. Effect of subcooling at 1000 psig on the under disc-to-upstream pressure ratio.

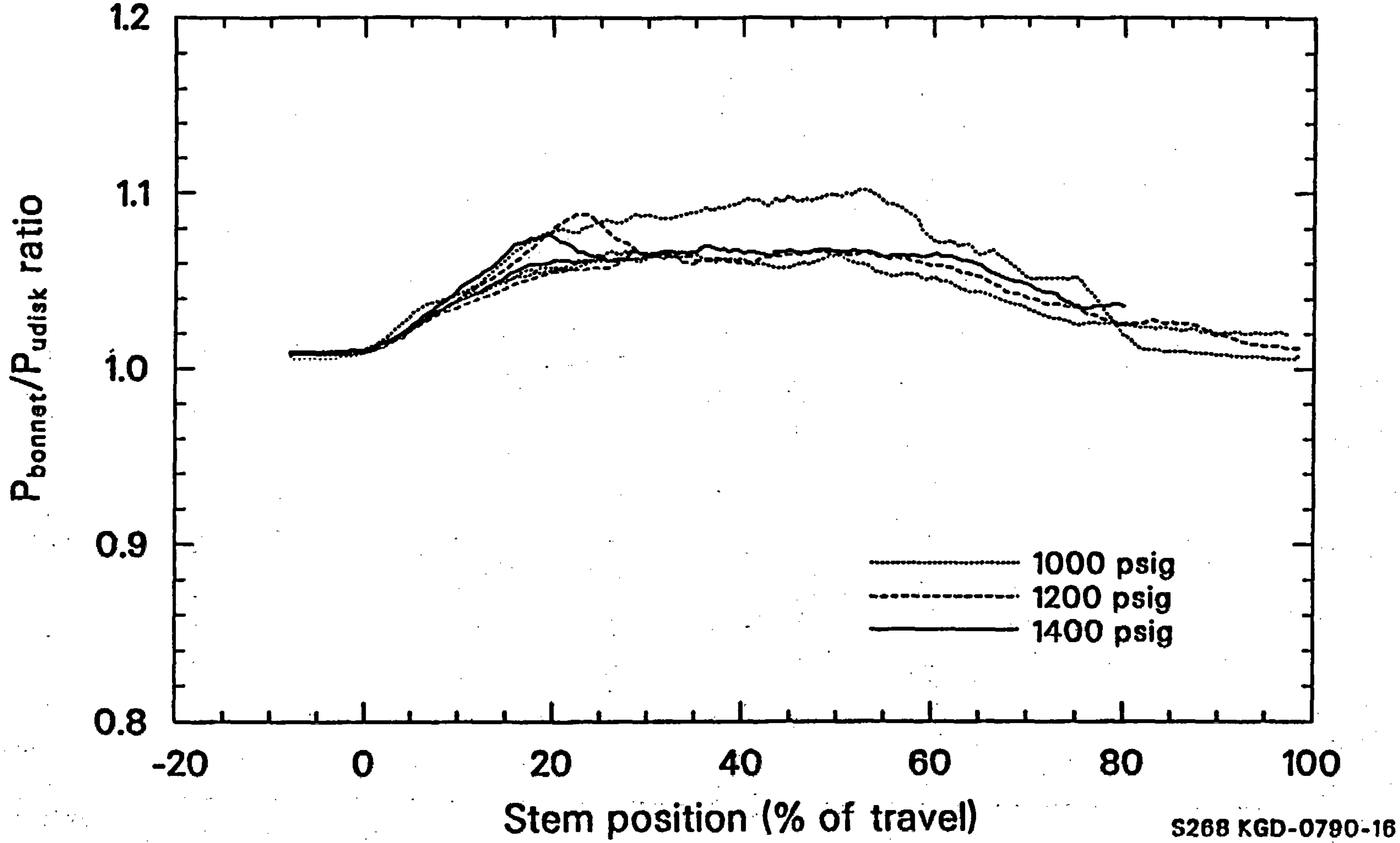

Figure 39. Valve 3 closing at full flow. Effect of pressure at 10°F subcooling on the bonnet-to-under disc pressure ratio.

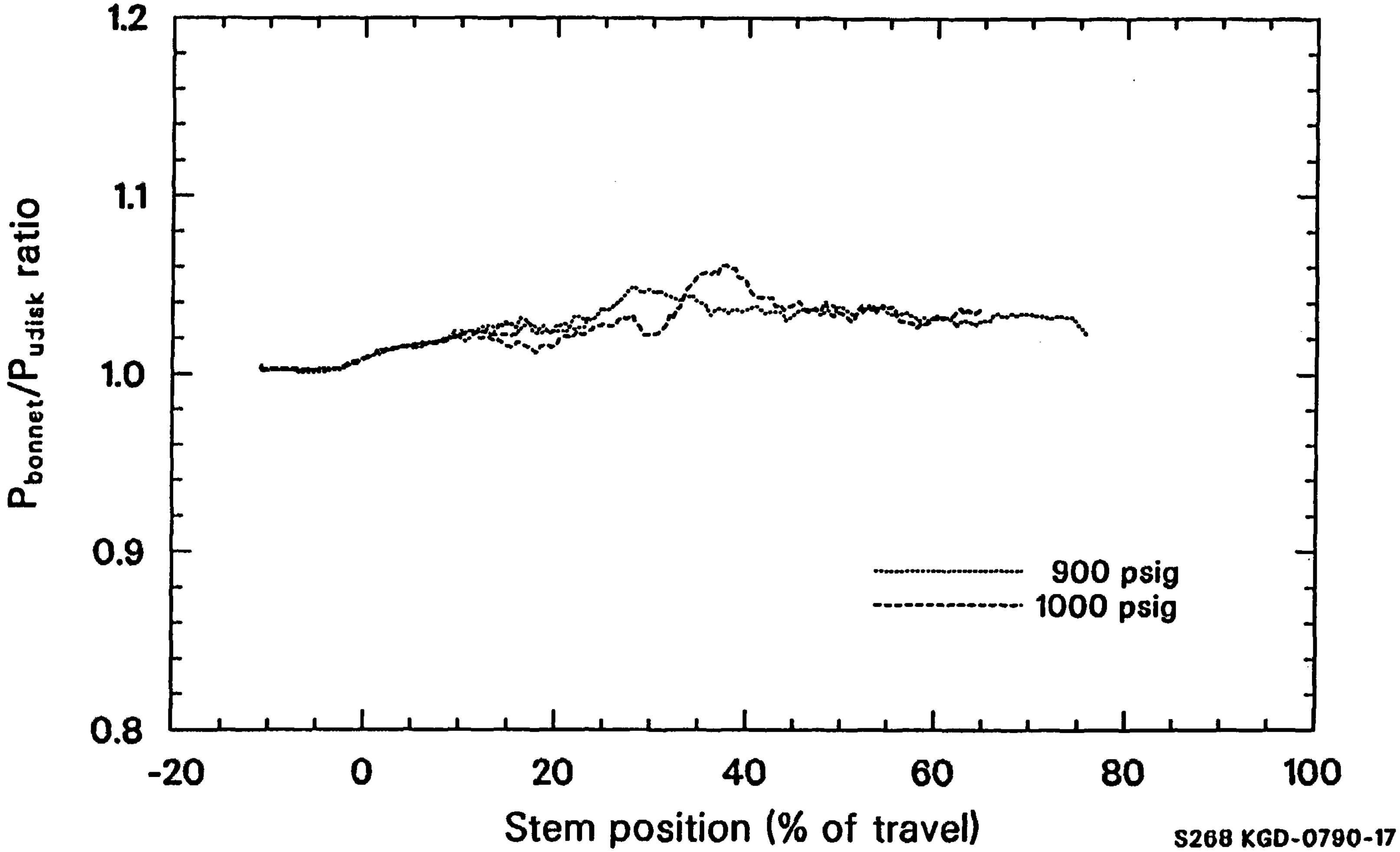

Figure 40. Valve 5 closing at full flow. Effect of pressure at saturation on the bonnet-to-under disc pressure ratio.

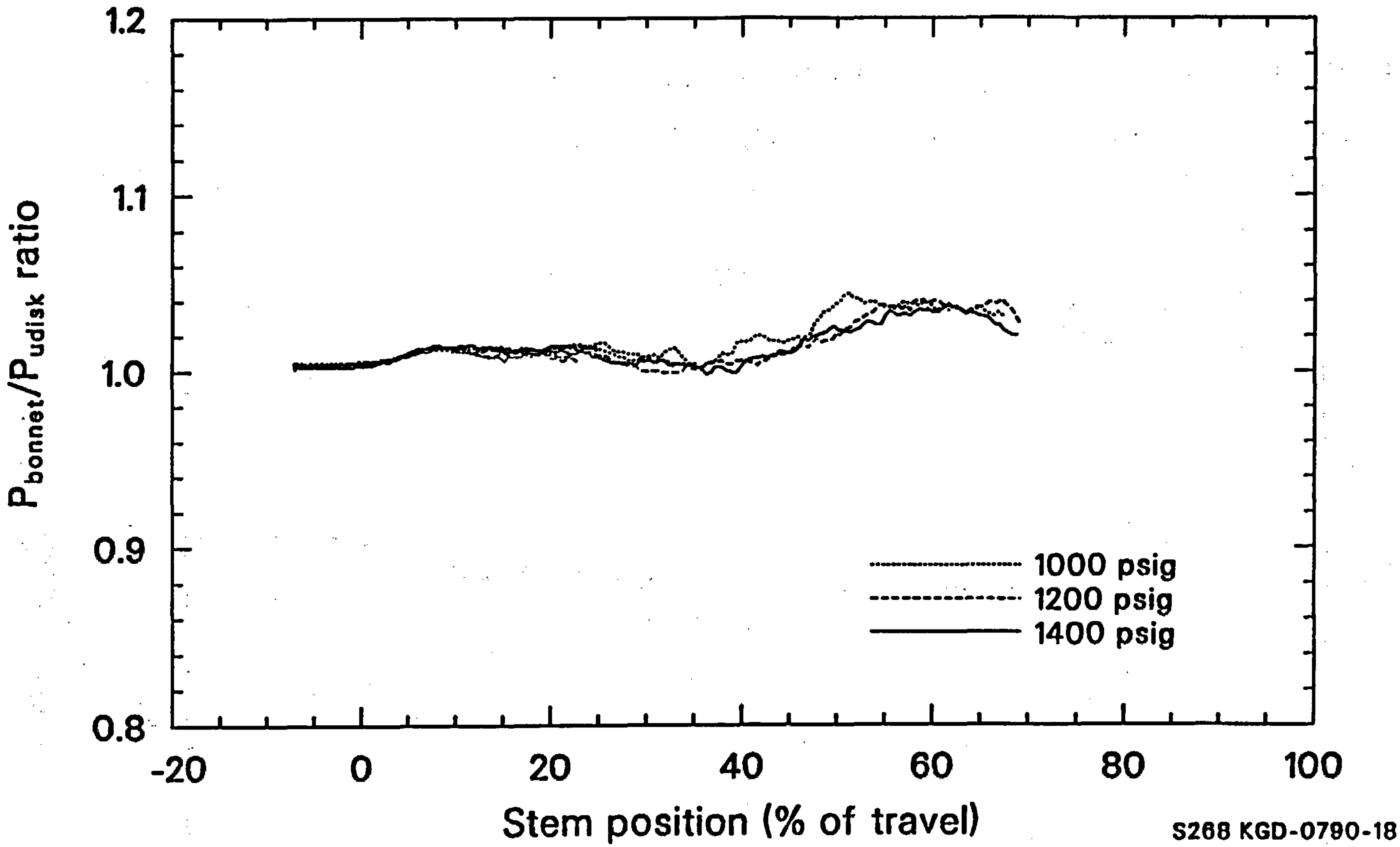

Figure 41. Valve 6 closing at full flow. Effect of pressure at saturation on the bonnet-to-under disc pressure ratio.

In addition to the GI-87-related test results described to this point, we have identified other concerns from the testing that may help the industry as they attempt to understand the concerns of GL 89-10 and GI-87.

4.4.1 Concern 1: Performing a Design Basis Test

Not all nuclear plant valves that open or close communicate with large reservoirs such as the primary or secondary system vessels. Some, however, must be capable of opening against large differential pressure loads or isolating potentially high pressure flows. Valves that can be tested in-situ at the worst-case loads, including the design basis pressure, temperature, and flow, are not a part of this concern once they have been successfully tested. These in-plant tests will determine the margin available in the operator and careful examination of the stem force, torque, and current and voltage plots should indicate possible damage from the loading.

Valves that must operate at potentially high loads and cannot be tested in place at those loads are the concern here. This includes valves that cannot be tested with the full design basis pressure, temperature, and flow loading over at least 60% of the full closure stroke. We have found that closing flexible wedge gate valves from less than this amount will not identify valve designs that are subject to nonpredictable performance. Thus, the actual stem forces required for closure cannot be detected.

Careful inspection of the valves after the GI-87 blowdown tests showed that valve damage started at about the two-thirds open point on the seats and guides. The buildup of material in the guides, or the depth of the seat machining at final closure, was influenced by this early damage. Early and late damage affected the maximum stem force at flow isolation. Measured stem forces at final seating, the last 5-10% of the closure, were less than at isolation in some cases.

Figure 10 contains the stem force history for Valve 4 during the simulated full-scale pipe break closure stroke. Note how the stem force history became very jagged when the valve was 40% closed. This jagged shape typically indicated damage to the valve during the flow isolation cycle, resulting in higher closure stem forces than would be expected from sliding friction. Also, note how the stem force decreased in Figure 10 during the last 15% of the closure. Figure 42 is a photograph of the damage to the valve guide rail observed during posttest inspection of Valve 4. Here we see the damage to the guide rail that began at about the 40% closed position and increased throughout closure to flow isolation. It is unlikely that a closure of only the last few percent would have caused the same damage and revealed the high closure forces necessary to obtain flow isolation at the design basis loading.

4.4.2 Concern 2: Motor Operator Dynamics (Rate of Loading)

The research program also found that measurements of torque, stem force, and motor performance are needed to completely characterize MOV performance. The measurement of torque or stem force alone will not identify problems in the conversion of torque to stem force (such as abnormal high stem factors) [Equation (2)]. At normal valve loading, torque and stem force data can be misleading because of motor-operator dynamic response (rate of loading). The rate of loading phenomenon is a highly visible issue in the industry today and its basis is not understood thoroughly. We believe that the effects of rate of loading are seen only when a valve does not seat fully enough to develop its potential seating stiffness. The best description of the problem is that the delivered stem force at torque switch trip is greater when the valve stem is loaded at a faster rate.

Limitorque informed the INEL that on rare occasions they have also seen the rate of loading effect on motor operators on their dynamometer testing. The difference in delivered torque was small. With the advent of in-situ diagnostic testing in the plants, there have been a number valves experiencing the rate of loading effect. This is typically found when they compare the difference in delivered stem force for a no load static test and a differential pressure test. Industry has developed many complicated analytical theories to explain the phenomenon. From our experience, we believe the most valid theory is the simplest one. If less stem force is obtained in the differential pressure test than in a no load static test, the valve application is operating in the margin between fully seating and some smaller degree of closure.

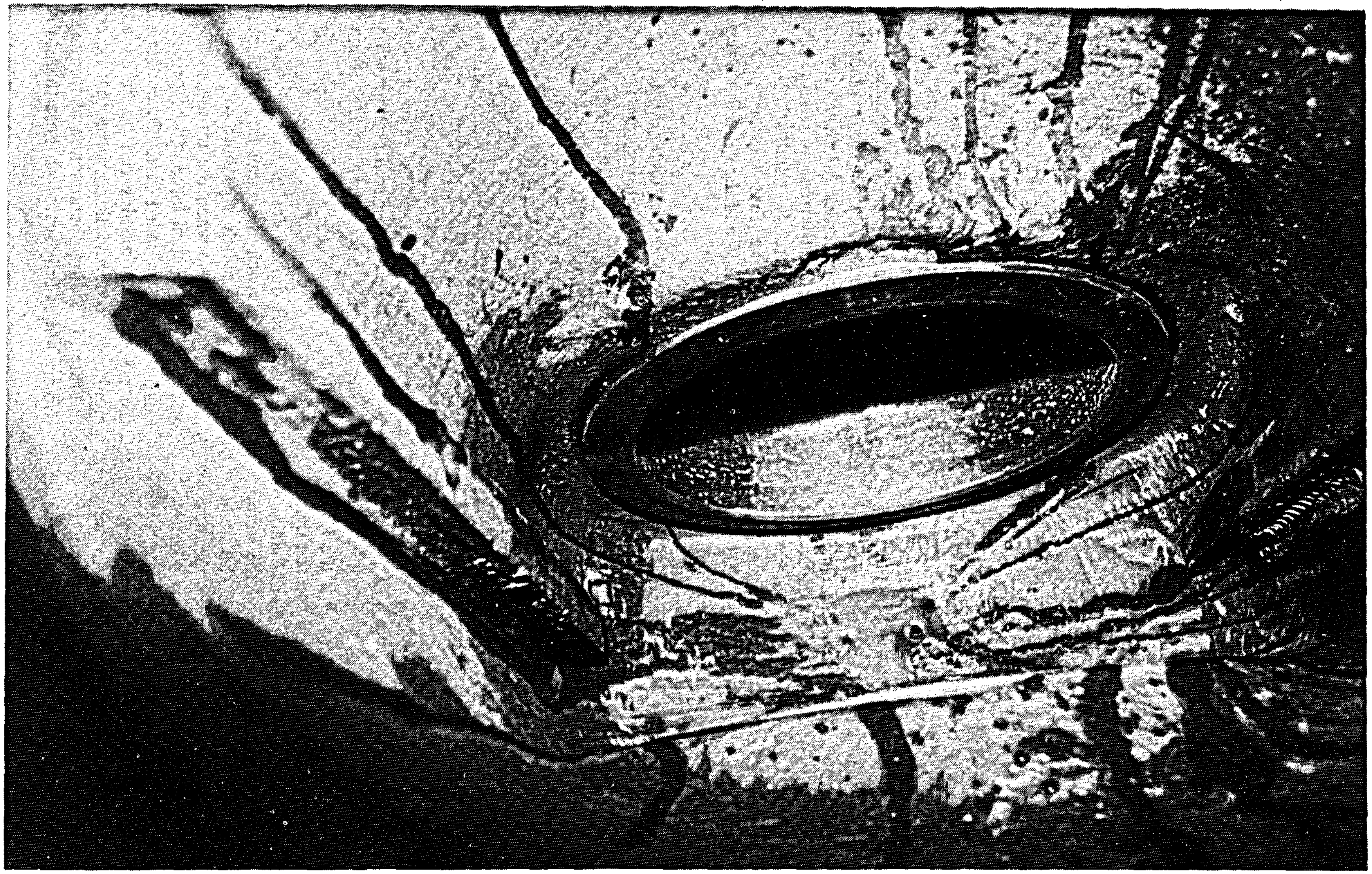

Figure 42. Damage to the seat and guide rail of Valve 4 because of closure loads.

We saw the effects of rate of loading twice in our test programs: once on Valve B during Phase I testing, and once on Valve 1 during Phase II testing. Both times, the rate of loading effect was the result of not setting the torque switch high enough to fully seat the valve under the differential pressure loading. In Figure 43, we see the stem forces measured as Valve B closed against three different pressures at pipe break flow in Phase I. Note that at the same torque switch setting, the stem force when the torque switch tripped in the 600-psig test with high flow is less than (18,100 lb) the stem force when the torque switch tripped in the no-flow static pressure test (19,900 lb) (see Figure 44).

The valve closing at 1000 psig shows a higher load before isolation of flow than in the 600-psig test. Just before this test, the valve stem was lubricated and a slightly higher stem force (18,600 lb) was obtained when the torque switch tripped. However, the valve stem position and the subsequent reopening of the valve indicated that the valve was lightly seated and the measured stem force was a greater reflection of closing load than of seating load. During the closing at 1400-psig inlet pressure (the design basis for this operator sizing and the torque switch setting), the valve marginally isolated flow but did not seat, and the operator tripped on disc load. The stem force when the torque switch tripped was lower (at 16,500 lb, a 17% reduction in the stem force at torque switch trip and a 25% reduction in final stem force) than with the lightly loaded case (Figure 43).

Figure 45 shows the same rate of loading comparison for Valve 6. Two stem force traces and two stem position traces are shown on the same plot. We have shown only the last 4.5 seconds of these full valve closures for clarity. The Step 30 trace is unloaded and Step 25 is the line break differential pressure test. You can see that in both of the stem force traces the rate of loading is equal prior to torque switch trip and the stem force at torque switch trip is the same. The stem position traces show a slight variation in operator momentum because of the loading but it is not significant. The time base for each trace was shifted slightly for clarity. This valve does not show the rate of loading effect. The valve seat was the resistance to motor operator momentum in both cases prior to torque switch trip and the valve achieved the same internal stiffness resulting in the same stem force.

To understand the rate of loading issue, one must visualize the two motion paths in the motor operator:

- The worm drives the worm gear, resulting in valve stem motion
- The worm climbs the worm gear, compressing the torque spring to torque switch trip.

The energy output from the electric motor takes the path of least resistance, and the fraction shared by each is determined by the relative resistance or stiffness of each path. When a valve is tested under low load conditions, the seat is the first significant load the motor-operator encounters. When this happens, the valve becomes a highly resistive or stiff assembly and the first motion path becomes essentially static. All the energy from the motor is then transferred to compressing the torque spring, virtually eliminating the losses associated with the first motion path. The assembly is very stiff, and the work input results in high stem forces.

When a valve is tested under high differential pressure conditions, which results in a high stem load, the spring pack is being compressed simultaneously with the rotation of the stem nut. Thus, the motor output energy is split between the two motion paths and their frictional losses. The additional inefficiencies can result in lower stem forces compared to the low load condition if the torque switch is tripped prior to achieving the same seating stiffness as in the unloaded case. Therefore, the relationship of motor-operator torque-to-valve stem force with both motion paths active can only be ensured under design basis loading.

If the loads during design basis closure can be overcome to reach the same structural stiffness prior to torque switch trip, there will only be one motion path active when the torque switch trips, and the resulting stem force will be primarily the same as in the unloaded case. Once determined, the no-load relationships can be baselined and monitored for degradation. Valves that experience the rate of loading phenomenon at torque switch trip are not as fully seated as those that do not.

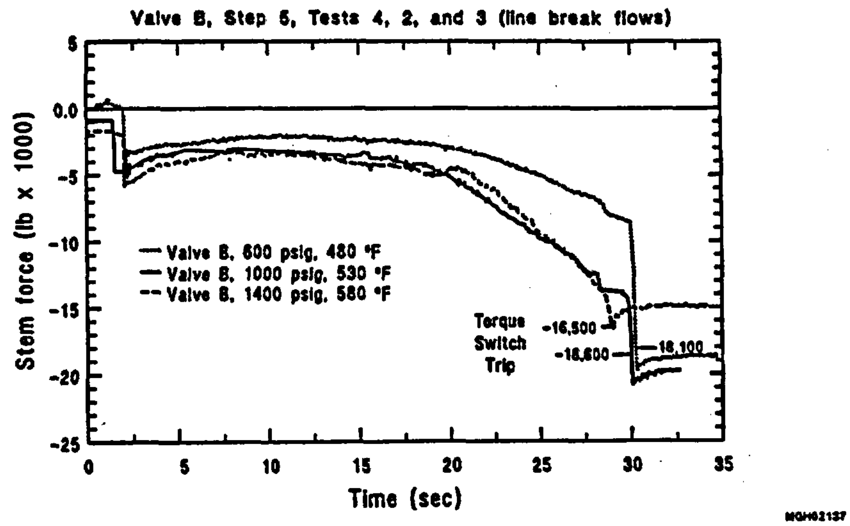

Figure 43. Valve B stem force measured at pipe break flow.

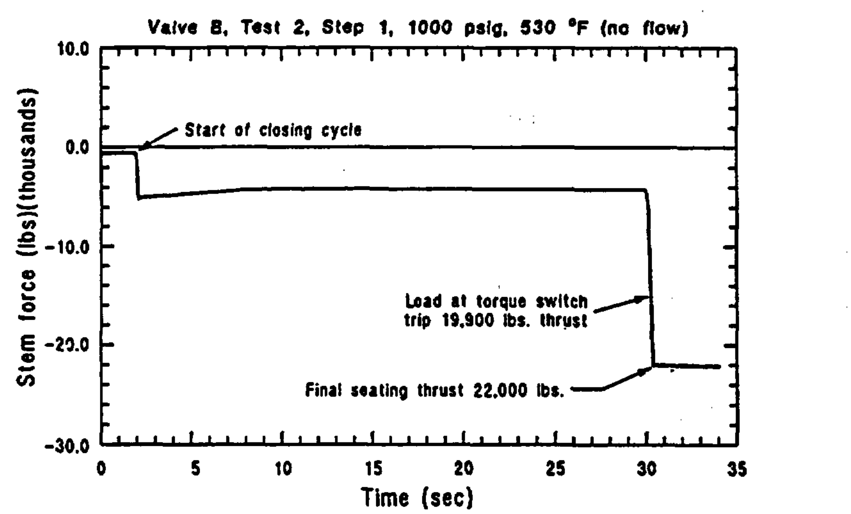

Figure 44. Valve B stem force measured at no flow.

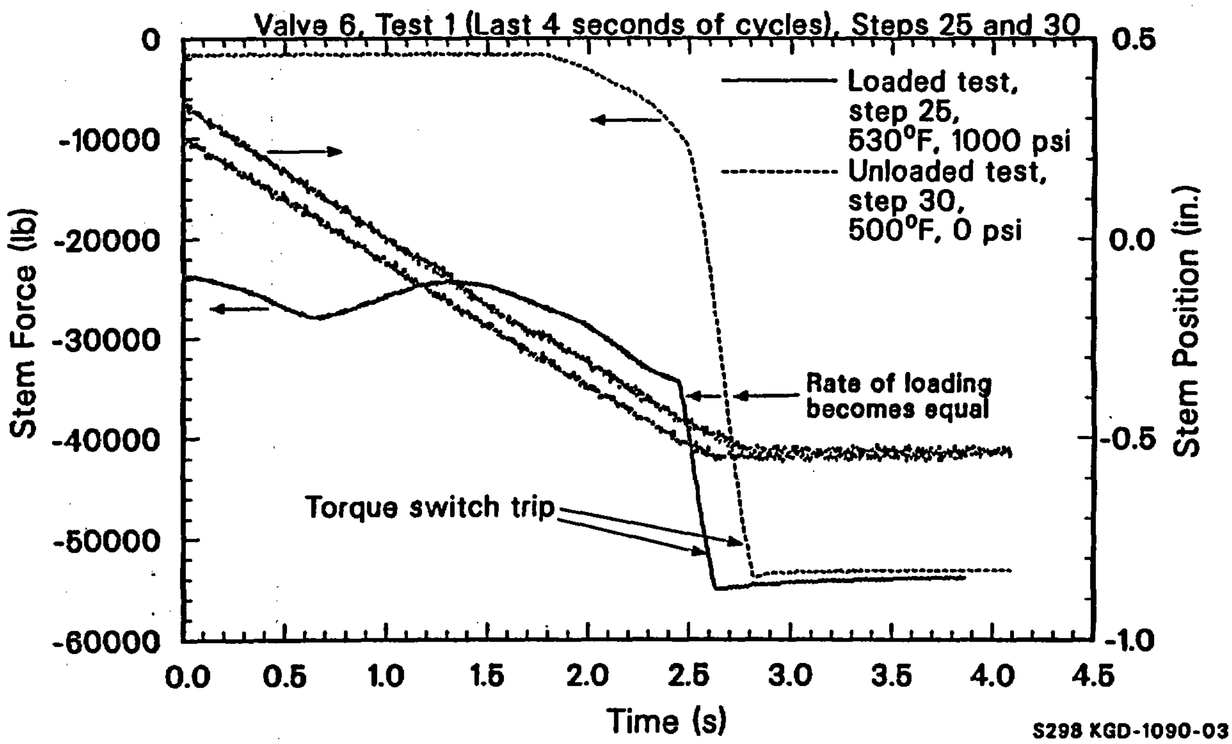

Figure 45. Phase II, 10-in. HPCI system valve, stem position and stem force at torque switch trip, comparison for a loaded and unloaded valve test.

These valves are operating in the margin between successful and unsuccessful closure. When possible, increasing the torque switch setting enough to fully seat the valve at design basis loadings will reduce or eliminate the rate of loading effect.

4.4.3 Concern 3: Critical Test Measurements

Our experience has shown that torque, stem force, and motor current and voltage should be measured during all valve tests in addition to motor operator switch position. As discussed above, torque and stem force measurements will provide the analyst with the stem factor (ratio of torque divided by stem force) for the valve. The motor current and voltage measurements allow the analyst to determine motor margins.

If the utilities could subject a valve to an in-situ design basis test every time testing is required by maintenance activities or regulation, they may not need to subject the valve to diagnostic testing at all (valves would have to be checked for damage). Routine design basis testing is not typically done in the plant. Instead, a utility may ideally subject a valve to prototypical or in-situ testing at design basis loads and then use diagnostic measurements to determine the force required to open or close it. This establishes the baseline data for the valve. Testing then could be done under static conditions (no pressure or flow in the line) to determine functional degradation.

Measuring both torque and stem force during the baseline differential pressure and static tests establishes the torque-to-stem force ratio (stem factor) for the valve. This is particularly important for valves that exhibit any rate of loading effects in the differential pressure test. These baseline values can then be compared to the calculated values for a given motor operator and valve assembly to determine the credibility of the measured values and to identify possible problems in the stem nut-to-valve stem conversion.

We recently were involved with design basis testing of a 4-in. gate valve, which was not part of the Phase I or II test program. The measurements of torque and stem force did not correspond, indicating a problem in the stem factor or in the credibility of the measurements. The 4-in. valve had a 1-in. diameter stem with 1/6 pitch and lead thread configuration. Using the Limitorque dynamometer-determined torque switch position versus output torque calibration and the power thread calculation equations, we constructed a calculated stem force-versus-torque switch position plot for various stem nut coefficients of friction (stem factors). Figure 46 is based on the Limitorque calibration of motor-operator output torque as a function of torque switch position. Using stem nut friction coefficients varying from 0.1 to a worst case of 0.2, we constructed the stem force versus torque curves shown in Figure 47. These curves allow one to determine what the stem/stem nut coefficient of friction is (within the 0.1 to 0.2 friction range) using both the actual torque and stem force measurements.

Figures 48 and 49 are the actual torque and stem force measurements, indicating 144-ft-lb torque and 9800-lb at torque switch trip. However, Figure 47 indicates the intersection of these two points to be off-scale with regard to the 0.1 to 0.2 friction range. The rate of loading shown in the torque and stem force plots and the overshoot in the torque plot, indicate that the valve was seated. Comparing the measured stem force with the calculated stem force indicates that either the stem nut had a very high coefficient of friction or that the torque or stem force measurement was in error. The torque measurement for a torque switch setting of 2 appears to be correct. The measured motor current was comparable to the Limitorque motor-operator calibration for a torque switch setting of 2.

A faulty stem force measurement without a reliable torque measurement could have led the analyst to erroneously recommend a higher torque switch setting, possibly overstressing the valve and/or operator components. (The sister valve to this one required a torque switch setting of 2.5 to obtain approximately the same stem force as the test valve with a setting of 2.0.) If the coefficient of friction had been excessive, we believe the problem in the valve could not have been discovered without both measurements.

4.4.4 Concern 4: Motor Performance Characteristics

In anticipation of higher stem force demands, Limitorque is in the process of requalifying some of their operators for greater stem force capacities. In some cases, utilities may need to set their torque switches higher and it will be necessary to determine if the valve structural capacity and the electric motor capability of the motor operator can take the higher loads. Typically in the past, the diagnostic motor current measurement has been made for trending purposes. Our research has shown that comparisons between the motor manufacturer's motor performance curves and measured current and voltage are very important for evaluating motor operating margins. Included in this are the calculations that should be made for degraded voltage concern and the length and size of the power cables. A small amount of motor or cable heat can have a large effect on a marginally sized motor's performance.

An example of the motor capacity problem is shown in Figures 50 and 51. These plots show two consecutive stem force traces for Valve 5, closing to isolate full pipe break flow at normal BWR HPCI operating pressure and temperature, then opening to 30% open, and reclosing to isolate full flow again. The time axis indicates that the two steps amount to about 38 s of highly loaded operation for this 15-min duty cycle motor. A look at the current traces (Figure 52) shows that during the first closure, the current is exceeding or has exceeded the 20-A maximum range of the transducer at torque switch trip. A "normal" trip was experienced, however. In the reopening and reclosing step (Figure 53), the current again exceeds the maximum range of the transducer at the end of the valve stroke, but the motor is never able to generate enough torque to reach torque switch trip, resulting in a stall as evidenced by the current transducer remaining saturated for about 4 s before we manually stopped the motor.

Figure 54, which shows the manufacturer's motor performance curves, gives some information about this behavior. The solid line indicates the expected performance of the motor, both motor speed and current-versus-motor torque. During the test, we observed that the voltage at the motor dropped from 460 V ac to as low as 380 V because of the response of the power supply, the resistance in the motor cables, and the stall or near stall currents. The dashed line in the motor speed plot shows the expected motor performance, corrected for the actual measured voltage. Actual motor performance observed during this test is shown by the dotted line, indicating that the motor was actually performing as anticipated at the measured voltage.

Unfortunately, we designed our current measuring system based on a Limitorque recommended 40-ft-lb motor. Limitorque later notified us that a 60-ft-lb motor would be necessary for Valve 5. If our measurement system could have handled the unexpected higher current reading from the larger motor, we could have checked the current demand after the design basis test. We then could have determined whether we were operating the motor so far out on the knee of the motor torque speed curve that there was little safety margin left in the motor, or that a voltage drop was driving the motor's poor performance.

In our case, both conditions occurred. The important thing to remember, however, is that the verification of the motor margins can prevent a failure of the valve to function at the design basis load condition. Valve, operator, and motor momentum will typically carry a marginal motor through a normal torque out at low or static loads.

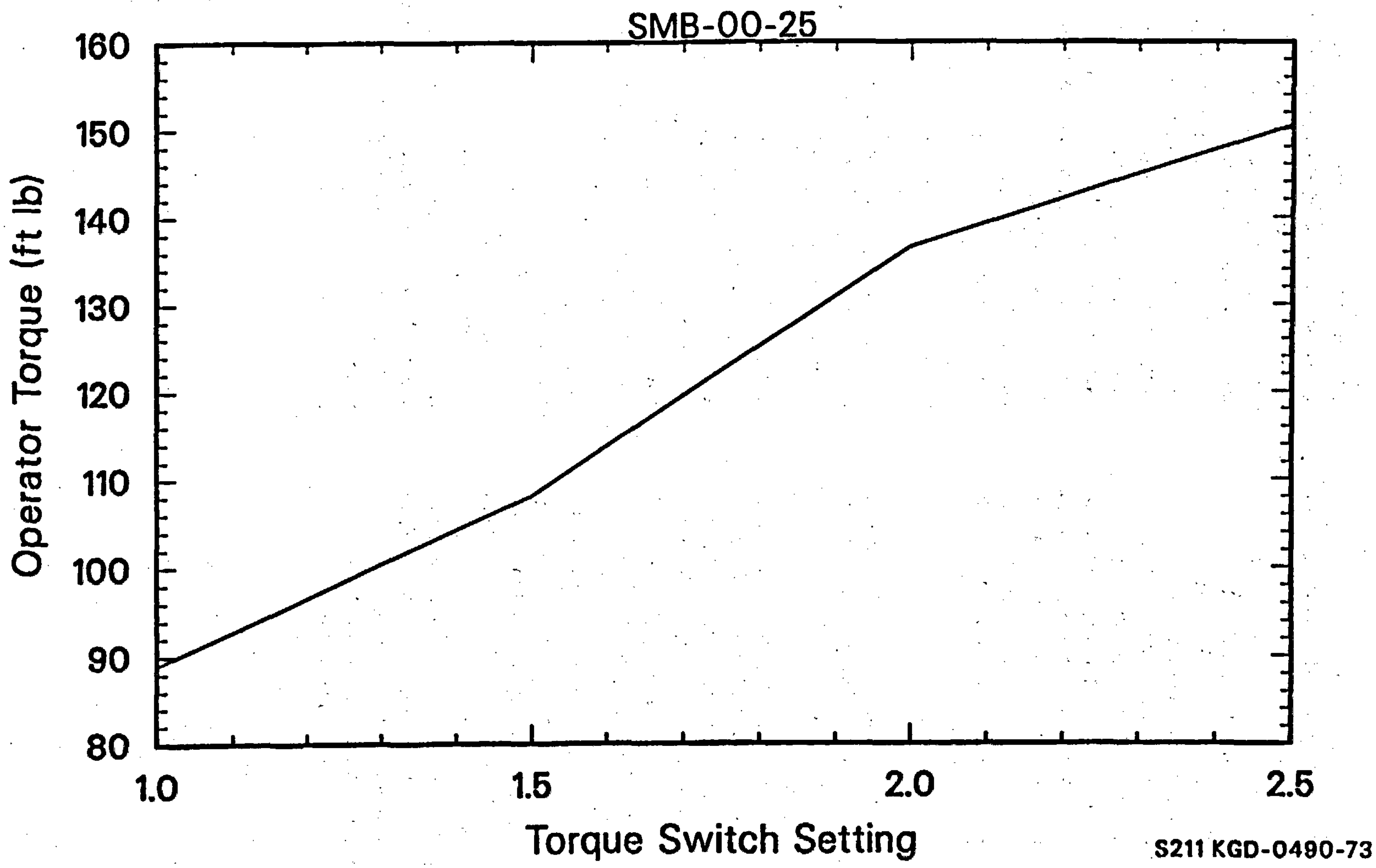

Figure 46. Limitorque calibration of motor operator output torque as a function of torque switch position for an SMB-00-25 operator.

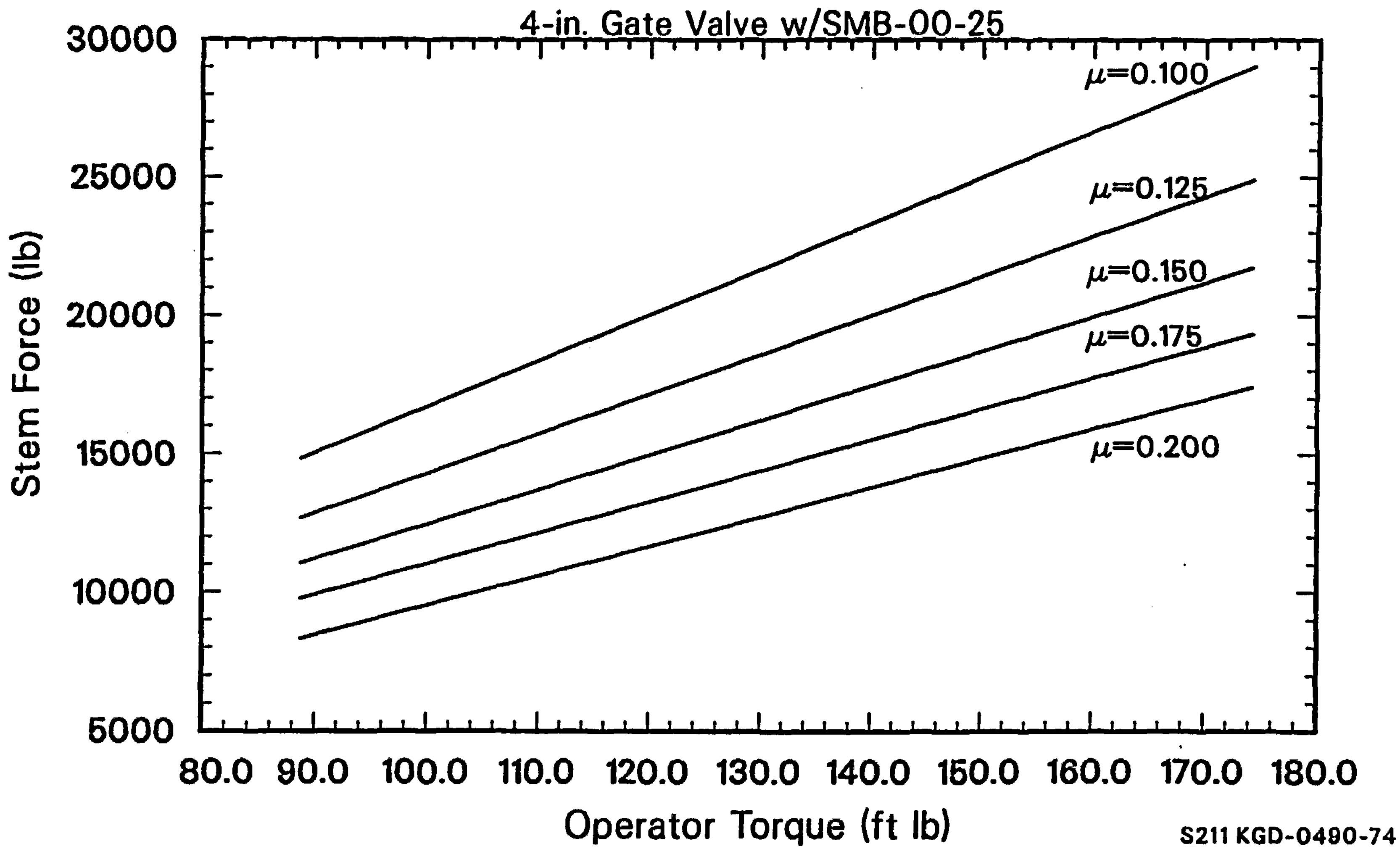

Figure 47. Comparison of stem force versus motor operator torque for various stem nut frictions.

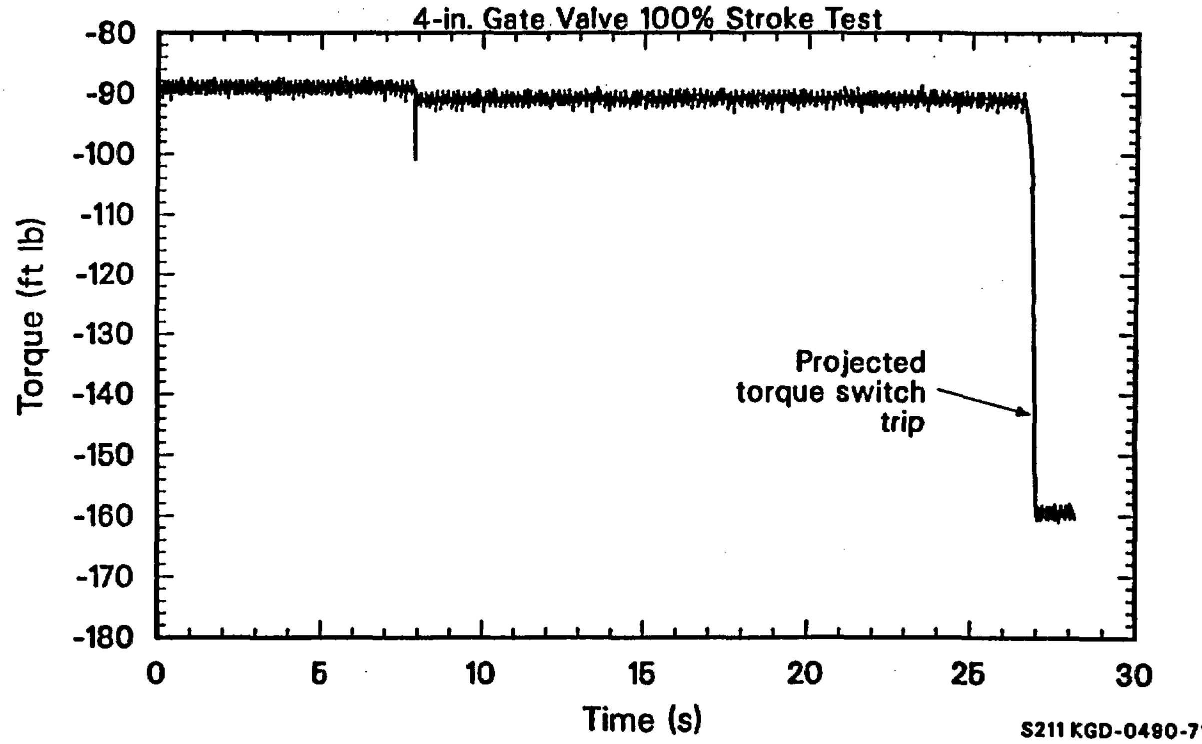

Figure 48. Torque measurements during a 4-in. valve design basis flow interruption test.

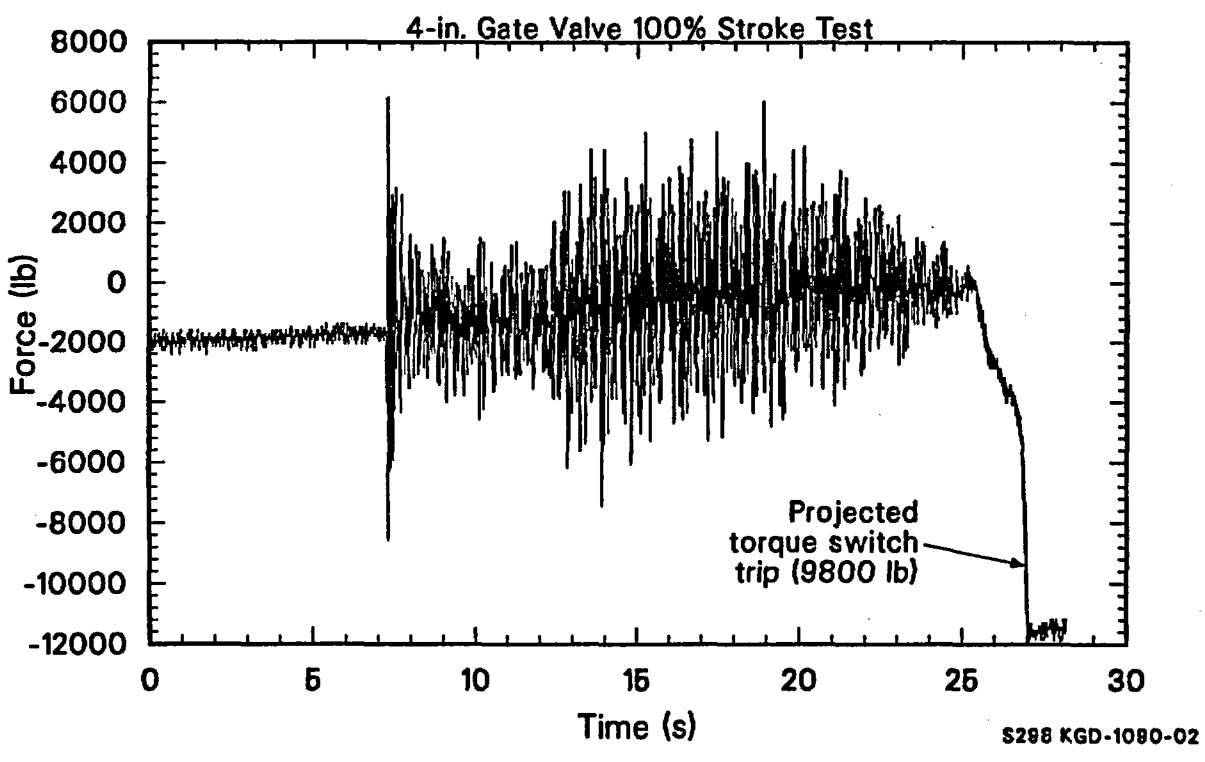

Figure 49. Force measurements during a 4-in. valve design basis flow interruption test.

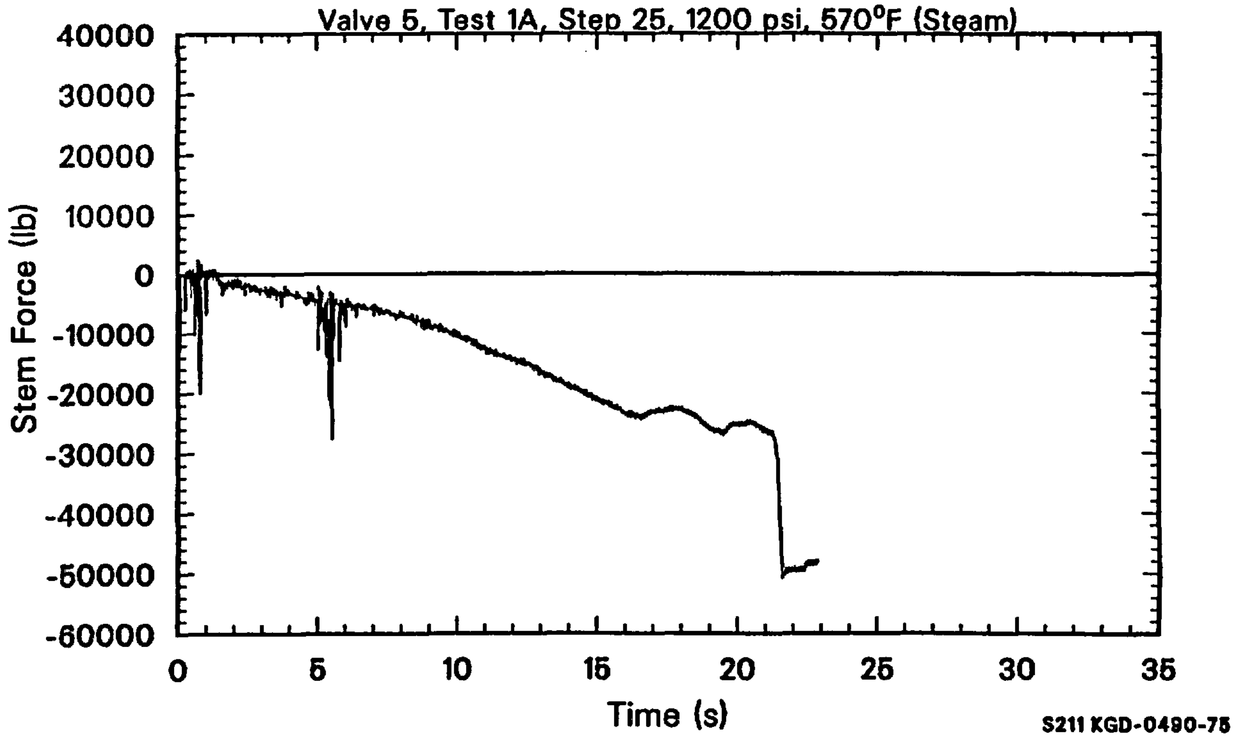

Figure 50. Stem force for Valve 5 closing at normal BWR HPCI operating pressure and temperature.

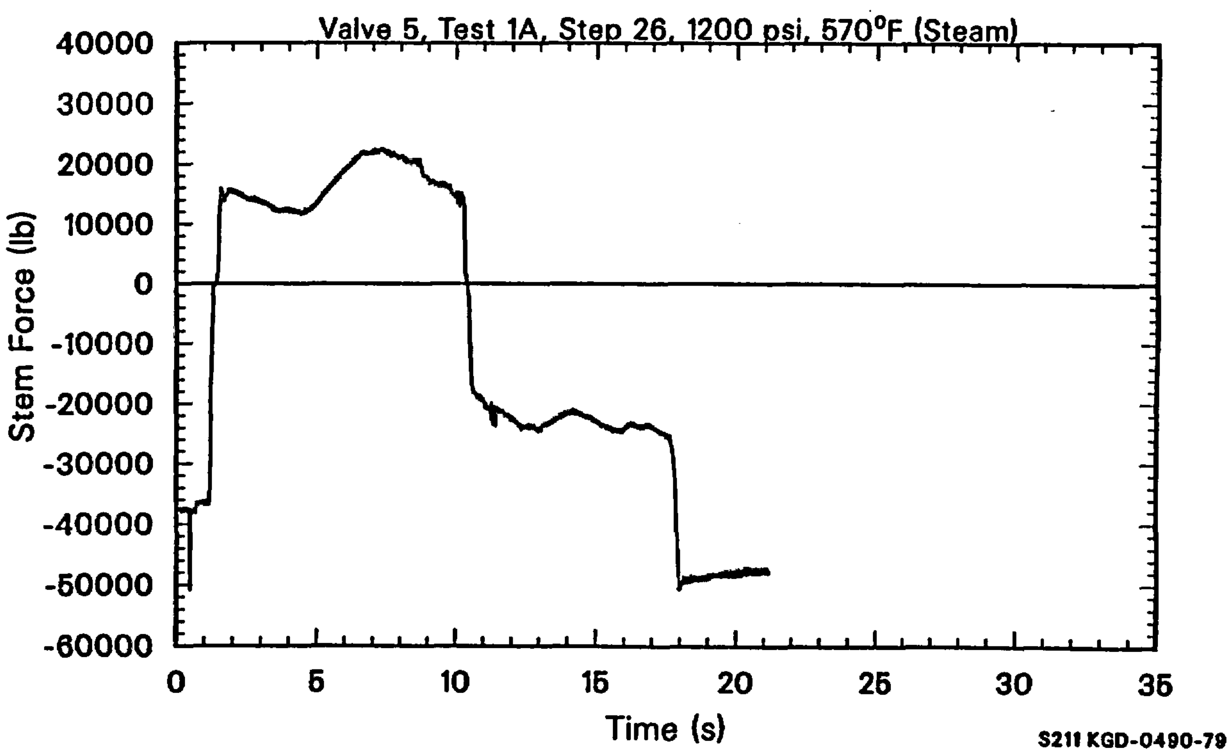

Figure 51. Stem force for Valve 5 opening to 30% open and reclosing.

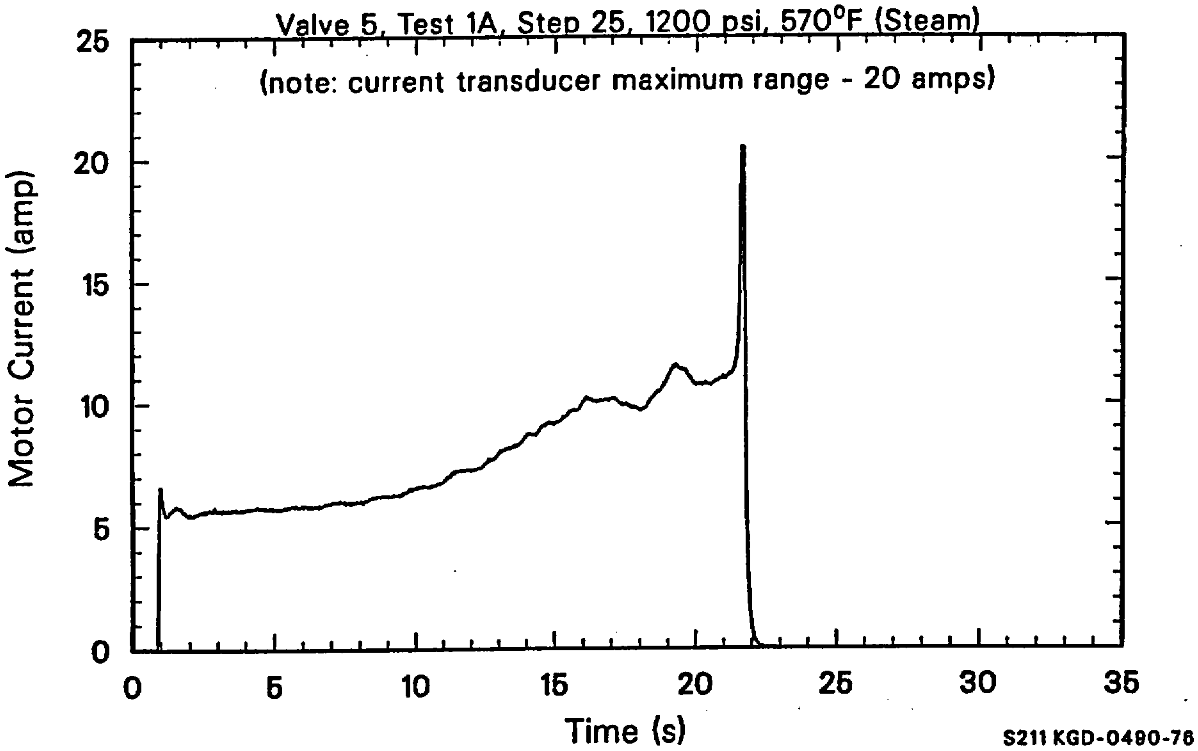

Figure 52. Motor current for Valve 5, closing at normal BWR HPCI operating pressure and temperature.

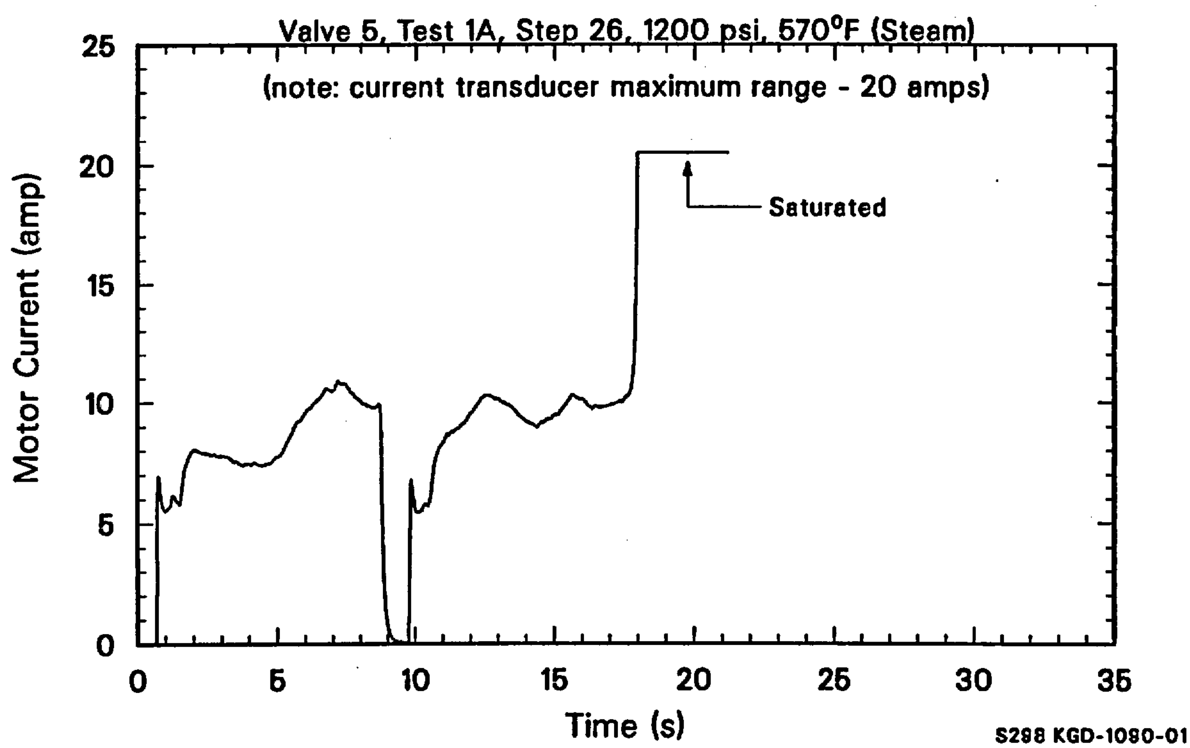

Figure 53. Motor current for Valve 5, opening to 30% open and reclosing.

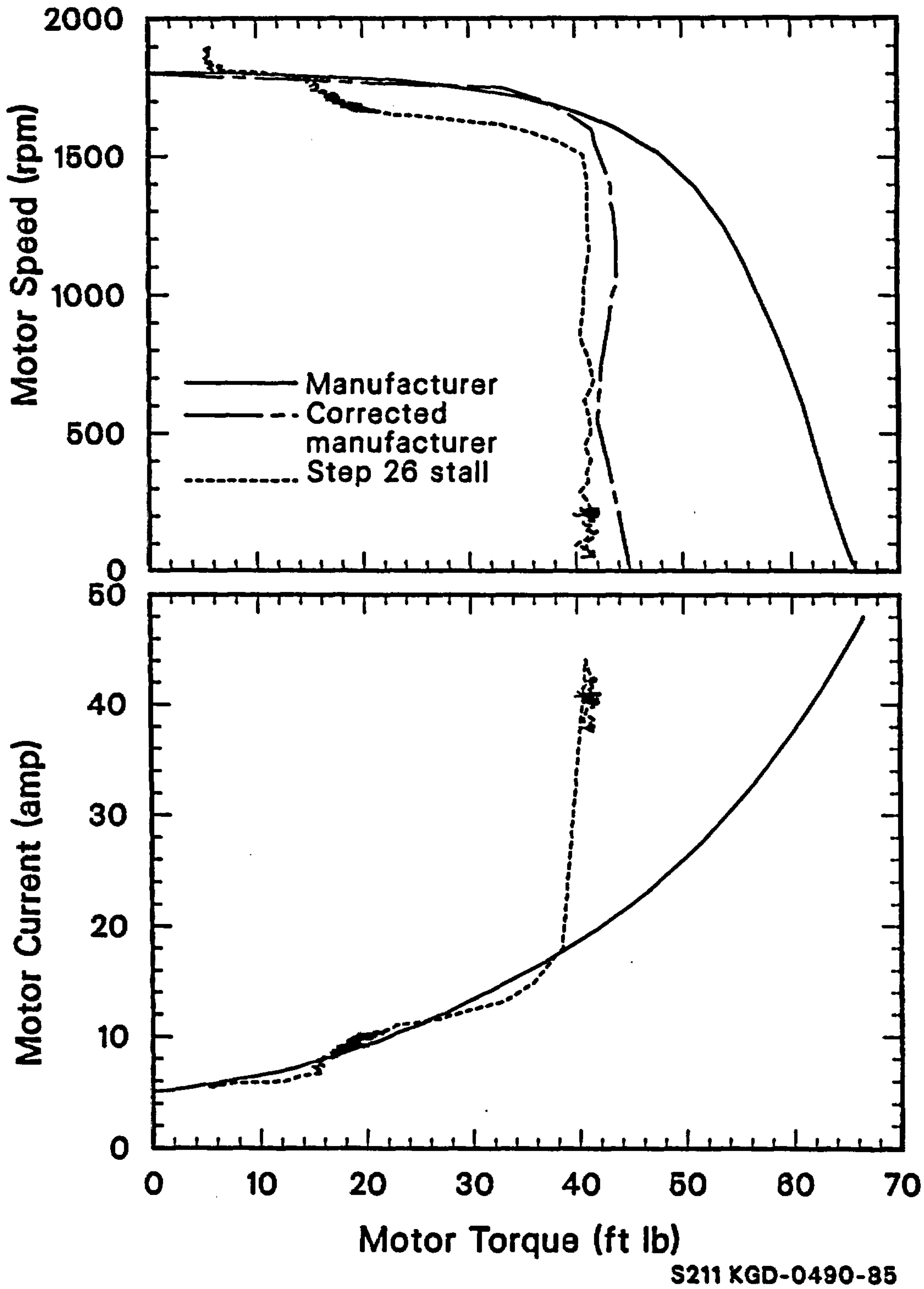

Figure 54. Comparison of Limitorque's motor performance curves and observed performance during motor trip.

5. CONCLUSIONS

The flexible wedge gate is the predominant valve design installed in the GI-87 systems of interest. Five manufacturers provided the largest fraction of these installed valves. Four of these manufacturers, which produce 90% of the installed valves, were represented in the GI-87 tests. The tested valves do not represent every variant of the manufacturer's valve product. However, they do represent the most popular designs with the exception of Valve 3, which had a flexible guide design. They differ slightly in manufacturing technique, but they are basically the same functional design. Valve 2 was tested in Phase I with a hardface disc guide and in Phase II with a normal disc guide. However, there was no discernible difference in performance between the two tests. Valve 6, the 10-in. version of 6-in. Valve 2, performed quite differently so it cannot be concluded that one manufacturer's product performs consistently across all sizes. Valve performance appears to be more valve-specific depending on stackup manufacturing tolerances, design clearances, and surface finishes and the way functional load affects the stackup and finishes. The valves with machined guiding surfaces had less guide damage than those with cast surfaces. That fact did not influence the frequency of seat damage, however.

Functionally, current industry models that are used to predict the MOV's performance are inadequate. Industry has performed very little testing on each design and apparently has lumped all of the unknowns into a disc friction coefficient. Facility limitations usually restrict industry testing to small valves. Test results from small valves obscure the internal pressure effects taking place and, because of the lower disc loadings, would not be as susceptible to the damage mechanisms we observed in the Phase I and Phase II tests.

All of the valves required more stem force and subsequently more torque to operate than what would have been estimated prior to the test program. This would indicate that even with the conservatisms that are added to typical motor-operator sizing calculations, there is a potential for undersized motor operators on the isolation valves of concern to GI-87.

Additionally, we believe that for the predictable valves used in this test series the actual disc sliding friction factor is probably in the 0.5 or 0.6 range at maximum differential pressure in both the opening and closing direction. Pressure distributions throughout the valve during opening and closing obscure the true value of friction being experienced in the valve. All of the loads that affect total disc load must be quantified for correct operator sizing.

Valves that sustained plastic deformation of the guide and sealing surfaces cannot be bounded by a disc factor equation that considers only sliding friction.

Valve diagnostic testing is in its infancy and many schemes are being offered to the nuclear utilities to respond to GL 89-10. The valve measurement system developed for this GI-87 testing has clearly identified a minimum diagnostic capability to ensure the credibility of the measurement and identify performance margins. At the time of the GI-87 test programs, none of the major diagnostic equipment suppliers provides this minimum diagnostic capability. Since the completion of the GI-87 test program, several suppliers have been working toward a more complete system. This will eventually help the utilities perform better tests.

The one significant problem that may not be solved easily deals with motor-operator electric motor performance data. Motor performance curves may not be obtainable for older motors and projecting accurate performance on more modern motors at higher loads may make it difficult for utilities to establish motor margins.

6. REFERENCES

ANSI/ASME (American National Standards Institute/American Society of Mechanical Engineers), 1983, B16.41, "Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants."

ANSI/ASME (American National Standards Institute/American Society of Mechanical Engineers), OM-8, "American Society of Mechanical Engineers Nuclear Codes and Standards Operation and Maintenance of Nuclear Power Plants, Section 8."

ANSI/ASME (American National Standards Institute/American Society of Mechanical Engineers), OM-10, "American Society of Mechanical Engineers Nuclear Codes and Standards Operation and Maintenance of Nuclear Power Plants, Section 10."

EG&G Idaho, 1990, *Generic Issue-87 Flexible Wedge Gate Valve Test Program Phase II Data Report*, Vols. 1-8, EGG-SSRE-8970, March 1.

EG&G Idaho, 1989, *BWR Reactor Water Cleanup System Flexible Wedge Gate Isolation Valve Qualification and High Energy Flow Interruption Test*, Vols. 1-3, NUREG/CR-5406, EGG-2569, October.

NRC, "Motor Operated Valve Common Mode Failures During Plant Transients Due to Improper Switch Settings," Office of Inspection and Enforcement, Bulletin 85-03.

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Qualification and flow isolation tests were conducted to analyze the ability of selected boiling water reactor process valves to perform their containment isolation functions at high energy pipe break conditions and other more normal flow conditions. Numerous parameters were measured to assess valve and motor-operator performance at various valve loadings and to assess industry practices for predicting valve and motor operator requirements. The valves tested were representative of those used in reactor water cleanup systems in boiling water reactors and those used in boiling water reactor high-pressure coolant injection (HPCI) steam lines. These tests will provide further information for the U.S. Nuclear Regulatory Commission Generic Issue-87, "Failure of the HPCI Steam Line Without Isolation," and Generic Letter 89-10, "Safety-related Motor Operated Valve Testing and Surveillance."

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